

Design of a water-CO₂ evaporator

Adoption of a carbon dioxide heat pump to
geothermal conditions

Mateusz Tatys



UNIVERSITY OF ICELAND



University
of Akureyri

VISIT...

LANZAROTE
Caliente.COM

DESIGN OF A WATER-CO₂ EVAPORATOR

Adoption of a carbon dioxide heat pump to geothermal conditions

Mateusz Tatys

A 30 credit units Master's thesis

Supervisor:

Dr. Ragnar Ásmundsson, ISOR, Iceland

A Master's thesis done at
RES | the School for Renewable Energy Science
in affiliation with
University of Iceland &
the University of Akureyri

Akureyri, February 2009

Design of a water-CO₂ evaporator

Adoption of a carbon dioxide heat pump to geothermal conditions

A 30 credit units Master's thesis

© Mateusz Tatys, 2009

RES | the School for Renewable Energy Science

Solborg at Nordurslod

IS600 Akureyri, Iceland

telephone: + 354 464 0100

www.res.is

Printed in 14/05/2009

at Stell Printing in Akureyri, Iceland

ABSTRACT

In this thesis the utilization of CO₂ heat pumps, together with low-enthalpy geothermal waters or energy of waste water, will be discussed and an attempt will be made to convince the reader that those devices can be used in a simple way to save energy and mitigate global warming.

Basing on the so-called EcoCute air to water CO₂ heat pump, a unit which can make use of low temperature geothermal resources, as the external lower heat source will be designed. To reach this target special attention will be paid to the design of a water to carbon dioxide evaporator.

The evaporator type suggested here is of a shell and tube design, with capillary copper tubes which are optimized for incoming water temperatures ranging from 15°C to 40°C and a working fluid pressure of 4MPa. This design will guarantee a small pressure drop on the CO₂ side, no leaks, relatively large heat transfer area and will be worth its price.

TABLE OF CONTENTS

1	Introduction.....	7
2	Heat pump theory.....	11
2.1	Reasons to use heat pumps	11
2.2	Principles of work.....	15
2.2.1	COP and other factors	15
2.2.2	Thermodynamic cycles	16
2.2.3	Absorption heat pumps.....	19
2.2.4	Refrigerants	20
2.3	Lower heat source	22
2.4	Upper heat source	24
3	CO ₂ heat pumps	25
3.1	Reasons to use.....	26
3.2	Construction.....	27
3.2.1	Compressor.....	27
3.2.2	Condenser.....	30
3.2.3	Casing dimensions.....	31
4	Design of the evaporator	34
4.1	Why is the switch of lower heat source from air to water so important?	34
4.2	Design assumptions	36
4.3	General idea – shell and tube heat exchanger.....	37
4.4	Design conditions – boundary conditions (temperature, pressure and enthalpy)	38
4.4.1	Water side.....	38
4.4.2	CO ₂ side.....	38
4.5	Mass flows	39
4.6	Capillary tubes	39
4.7	Flow division	40
4.8	Pressure drop	41
4.8.1	Major losses.....	41
4.8.2	Minor losses	42
4.8.3	Pressure losses - conclusions.....	44
4.9	Length of the heat exchanger.....	44
4.9.1	Length of the evaporator	45
4.9.2	Length of the superheater	53

4.9.3 Final design	54
4.10 Water mass flow at different LHS temperatures	54
4.11 Influence of the new design on COP	55
5 Conclusions.....	56
References	59
Appendix A	1
5.1 Table A - Properties of CO ₂ in temperature range 4°C - 37°C and pressure 38 bar ..	1
5.1.1 Part 1	1
5.1.2 Part 2	2
5.2 Table B - Properties of water in temperature range 15°C - 40°C and pressure 2 bar	4
5.2.1 Part 1	4
5.2.2 Part 2	6
6 Appendix B.....	1
6.1 Approach.....	1
6.2 Assumptions	1
6.3 Design temperature	2
6.3.1 Surface conductances	3
6.4 Losses through doors and windows	3
6.5 Losses through walls.....	3
6.6 Losses through balconies	5
6.7 Ventilation losses.....	6

LIST OF FIGURES

Figure 1 The Keeling Curve, shows atmospheric carbon dioxide as directly measured at Mauna Loa, Hawaii (NOAA, 2009)	7
Figure 2 Shares of global final energy consumption and CO ₂ emissions by sector, 2005. Source: (IEA, 2007).....	9
Figure 3 Household energy use by end-use. Source: (IEA, 2007)	9
Figure 4 A schematic of the heat pump installed in Grýtubakki, NE-Iceland.....	11
Figure 5 Comparison of energy efficiency between combustion-based equipment and a heat pump	12
Figure 6 Comparison of costs and CO ₂ emissions between heat pumps and commonly used heat sources for space heating (The Fuel Effect, 2008)	13
Figure 7 Distribution of the oil resources (BP, 2004)	14
Figure 8 Energy price trends (1987-2005) (WEC, 2008).....	15
Figure 9 A basic schematic figure of a heat pump	16
Figure 10 The reversed Carnot cycle presented in T-s diagram for working fluid R-22	17
Figure 11 A relationship between COP and temperature difference between a lower heat source and an upper heat source	18
Figure 12 The ideal vapor-compression refrigerant cycle presented in T-s diagram for the working fluid R-22	18
Figure 13 An approximation of the actual process presented in T-s diagram for working fluid R-22.....	19
Figure 14 A basic schematic of an absorption heat pump	20
Figure 15 Types of the most often installed open systems.....	23
Figure 16 Types of the most often installed close loops systems.....	23
Figure 17 Commercially available EcoCute models made by different companies (Hashimoto, 2006).....	26
Figure 18 Shipment of EcoCute units (Hashimoto, 2006)	26
Figure 19 A cutaway of two-stage rotary compressor (SANYO, 2006)	28
Figure 20 An approximation of actual cycle for EcoCute heat pump presented on T-s diagram.	29
Figure 21 The condenser installed in the EcoCute heat pump (Hashimoto, 2006).....	31
Figure 22 Dimensions of Sanyo EcoCute heat pump. Model SHP-C45DEN (SANYO, 2006).....	32
Figure 23 New casing dimensions.....	33
Figure 24 A relationship between COP and ambient temperature of the air to water EcoCute unit	34

Figure 25 Chart showing measurements done in Grýtubakki, before and after the installation of the heat exchanger in front of the evaporator	35
Figure 26 Chart showing power consumed by electric heaters. It decreased after the installment of the heat exchanger in front of the evaporator	36
Figure 27 Schema of the heat exchanger showing an imaginary division for two parts: the evaporator and the superheater. Numbers are used as lower indexes and they indicate boundary points in the heat exchanger.	37
Figure 28 Y-junction	42
Figure 29 A continuous cross section area reduction	43
Figure 30 A continuous cross section area widening	44
Figure 31 Resistance concept for a cylindrical tube.....	46
Figure 32 Parallel and counter flow heat exchanger	47
Figure 33 Two phase patterns in horizontal tubes.....	48
Figure 34 Convection coefficient values in the two phase region for different qualities....	51
Figure 35 Tube banks in-line arrangement.....	52
Figure 36 Chart showing how water mass flow depend on ground water temperature	55
Figure 37 Chart showing how COP depend on ground water temperature.....	56
Figure 38 Temperatures in Reykjavik during one year, sorted in a decreasing order (Valdimarsson, 2008)	2
Figure 39 Surface conductances (ASHRAE, 1993)	3
Figure 40 Parallel and in series thermal resistances arrangement	4
Figure 41 An example of parallel-series arrangement.....	5
Figure 42 Program for calculations of the heat exchanger	1

LIST OF TABLES

Table 1 Classification of commonly used working fluids	21
Table 2 The most often used lower heat sources – temperatures and properties	24
Table 3 Properties of carbon dioxide for given pressure 3.9 MPa and quality 4%	29
Table 4 Saturation temperature for given pressure 1.3 MPa	30
Table 5 Table shows independence of enthalpy on pressure.....	38
Table 6 Thermal conductivities of chosen materials	39
Table 7 Friction factors for symmetrical y-junctions (VDI, 1991)	42
Table 8 Parameters for saturation pressure equal to 3.85 MPa	45
Table 9 Properties of points with quality 0 and 1 for pressure 3.85 MPa	49
Table 10 Convection coefficient values in the two phase region for different qualities	50
Table 11 Dependence of COP and water mass flow on inlet water temperature	55
Table A Properties of CO ₂ in temperature range 4°C - 37°C and pressure 38 bar.....	A - 1
Table B Properties of water in temperature range 15°C - 40°C and pressure 2 bar.....	A - 4

1 INTRODUCTION

It is clear that fossil fuels are not infinite and eventually oil and gas resources will run out. Predictions differ, but most of them assume that easily accessible oil will disappear in the next 40 years and gas in the next 150 years (BP , 2008). Coal reserves are more abundant and could run power plants for the next 410 years (BP , 2008); however, a question arises: Should the world's oil, gas and coal reserves be burned?

Global surface temperatures increased $0.74 \pm 0.18^{\circ}\text{C}$ during the last 100 years (ending in 2005) (IPCC, 2007). The rise of carbon dioxide concentration in the atmosphere is considered to be responsible for most of the global warming. As shown on the Keeling Curve (figure 1), the mean CO_2 concentration in the atmosphere increased from about 315 ppmv (parts per million by volume) in 1958 to 385 ppmv in January 2009 (NOAA, 2009). According to Danny Harvey (Professor at Toronto University, specialist in climate change and global warming), a concentration of CO_2 at the level of 450 ppmv (doubling of preindustrial era) is critical and can cause a rise in the global mean surface temperature from 1°C to 5°C (Harvey, 2009). Even a one degree increase in temperature will cause irreversible effects, such as changes in oceanic chemistry and impact on marine life. This is alarming, but a 4°C increase would have catastrophic consequences including melting of Greenland's and the West Antarctic ice sheets, which would lead to sea level rising by 10 m and possibly stopping the gulfstream (Harvey, 2009). This could turn out to be the second biggest disaster that could possibly occur, just after total nuclear war. Global warming cannot be stopped, but its effects can be mitigated. To be efficient, action must start now.

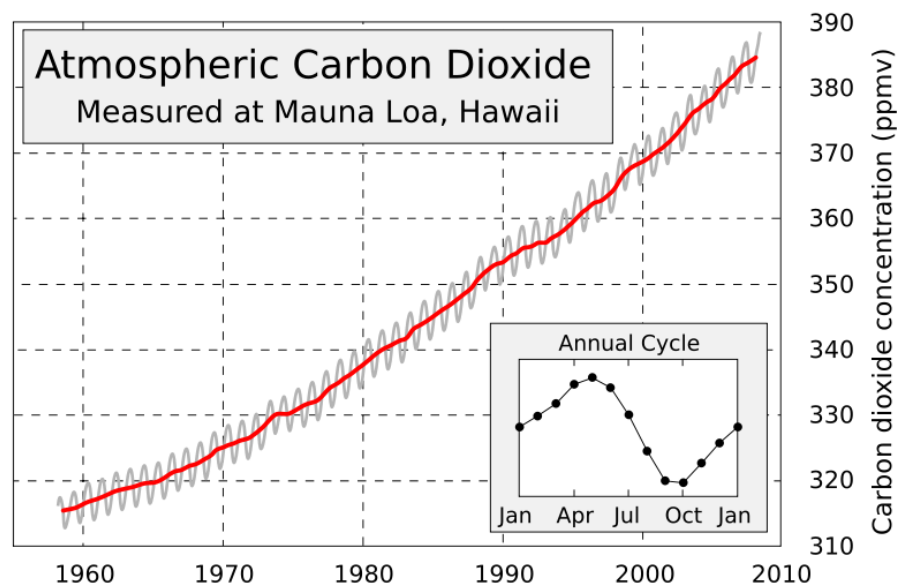


Figure 1 The Keeling Curve, shows atmospheric carbon dioxide as directly measured at Mauna Loa, Hawaii (NOAA, 2009)

The first steps towards mitigation have already been made. On 11 December 1997 in Japan, the Kyoto Protocol was signed, in which several countries agreed to reduce their CO₂ emissions. Until now the only significant emitter which hasn't ratified the protocol is the USA, but this is expected to change in the near future, since the country's Democratic Party supports the fight against climate changes. Worldwide emissions limits and the possibility to buy or sell those emissions will be the symbolic beginning of a new era in environmental protection. This would start the transition to less pollution, but potentially more expensive sources of energy and could be a technological boost for almost all countries.

Between 1990 and 2005 global energy use increased by 23%, while the associated CO₂ emissions rose by 25% (IEA, 2008). It is certain that the world's demand for energy will grow, and the challenge is to separate this trend from the increase in CO₂ emissions. However, when one looks closer at the future prognosis, this seems to be one of the greatest efforts of humankind.

The International Energy Agency (IEA) projects 8.1 billion earth inhabitants by 2030 (IEA, 2006), but some scenarios that extend to 2050 assume stabilization of human population at the level of 9 billion (Dutch/Shell, 2005). Gross Domestic Product will grow on average 3.4% per year to 2030 (IEA, 2006). It is important that India, China and other developing countries are expected to grow faster than the average. This will lead to an increase in the automotive market in these parts of the world and higher demand for liquid fuels, particularly oil. Taking into account that the world's demand for oil is projected to increase by 35% over 2005 levels by 2030 (DOE, 2008), production is supposed to rise by a similar factor, but what would happen if production capacity reaches its maximum? After the beginning of the current economic crisis, which started in the year 2008, many of these predictions will have to be revised, but it is irrefutable that eventually the availability of oil will expire. The situation when oil production rates cannot be increased is called peak oil. Optimistic estimations forecast the global decline will begin by 2020 or later (CERA, 2006), but pessimistic predictions assume that the peak has already occurred (Werner Zittel, Jörg Schindler, 2007), or that it will occur shortly (Koppelaar, 2005).

The shortage between demand and production of oil, limits on CO₂ emissions, growing demand for energy (because of increase in standard of life) in developing countries and the rise of global population will force us to look for new sources of energy. Technologies are emerging, but they are expensive or are still being developed (like nuclear fusion). Without governmental support, ambitious CO₂ emission reduction targets will be difficult, if not impossible. Implementation of renewable and nuclear power plants on a large scale will be time consuming. It seems that the easiest and fastest way to cope with growing energy demand without decreasing the life standard could be moderation of demand by increasing energy efficiency. What does it mean for one of the most important energy sectors - space heating? Is there any potential to reduce this energy demand?

Shares of Global Final Energy Consumption and CO₂ Emissions by Sector, 2005

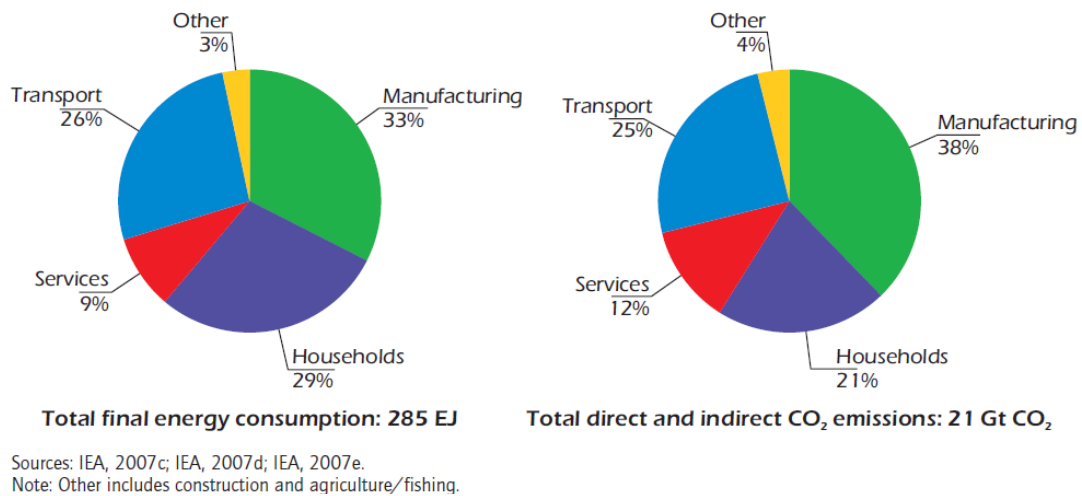


Figure 2 Shares of global final energy consumption and CO₂ emissions by sector, 2005.
Source: (IEA, 2007)

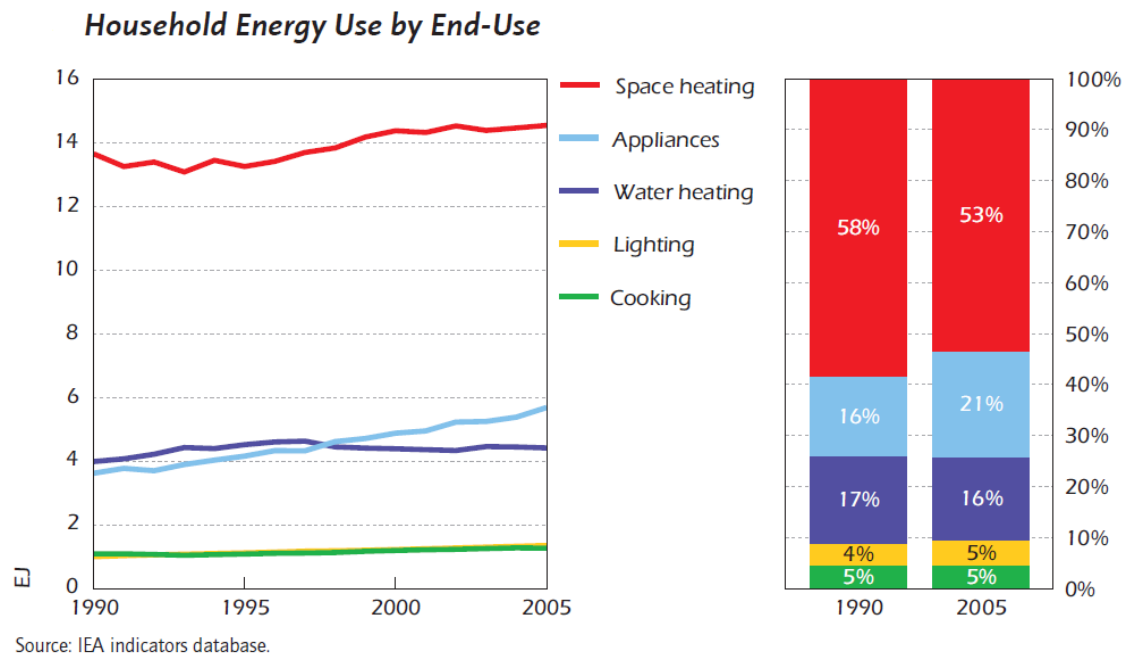


Figure 3 Household energy use by end-use. Source: (IEA, 2007)

It seems there is. In figure 2 we see that the housing sector is one of the three most important in global final energy consumption (29%) and CO₂ emissions (21%). According to IEA, space heating and domestic hot water represent 69% of total energy delivered to private householders in developed countries (figure 3). In industrialized countries this number might be different (for example the German electricity association VDEW claims it is even 87% in Germany (Ochsner, 2008)), but by far the largest share of energy consumed is by private householders. There are two ways to reduce costs and emissions in the space heating sector. The first way is to improve the isolation of buildings. This should

be connected with the implementation of new materials and promotion of passive house building techniques. Another way is to change the source of heat for a house. Of course those two approaches can be combined to achieve better results. Changing heat source means switching from conventional combustion based equipment to devices which take heat from the surroundings of a building. There are many such devices, but almost all are of limited use and not universally applicable. The reason might be price, too little power, an intermittent way of working or other factors. The best solution seems to be heat pumps, which take approximately 3/4 of their energy from their surroundings (they are described in the chapter 1). Until the end of the twentieth century the only problem with those devices were working fluids, which had high global warming potential (GWP). The situation changed when scientists and engineers considered natural working fluids, of which the most promising seems to be carbon dioxide (GWP=1 and good thermodynamic properties). The second step was done in 2001 in Japan, where an air-CO₂-water heat pump was developed and named the EcoCute unit (see chapter 3).

The goal of this thesis is to adapt this device for conditions in which relatively hot water is available as a lower heat source - LHS. Such European countries as Iceland, Hungary or Turkey have large amounts of low enthalpy geothermal waters with temperature ranging from 15°C up to 40°C, which could be utilized to a much greater extent. Underground geothermal reservoirs are not the only LHS to consider when designing and making a new evaporator; waste water within the same temperature range is also potentially viable and available all over the world.

In Iceland some steps have already been taken to utilize low temperature geothermal sources, and one example of a CO₂ heat pump installation exists. In 2006 and 2007, ISOR (Iceland GeoSurvey – Íslenskar orkurannsóknir) helped in the selection of the CO₂ heat pump and advised on its installation in Grýtubakki, NE-Iceland. The actual installation was performed by the land owner and certified electricians and plumbers. The heat pump's original design is based on the Japanese EcoCute (with some roots in Norway) and adapted for 'Nordic' conditions in Sweden, by a company called Ahlsell. The pump owner, Jonas Baldursson at Grytubakka, Eyjafjörður, added a simple air-water heat exchanger in front of the CO₂ heat pump inlet (figure 4). Water used in this heat exchanger has a temperature of 18-20°C and is taken from a borehole near the house. The exchanger improved the overall efficiency; air temperature at the entrance to the heat pump is higher and more stable. Nevertheless, in this solution only 31% of the thermal energy contained in the water is transferred to the evaporator (Ásmundsson, 2008). To improve this system a new evaporator design is presented in this thesis, in which water will exchange energy directly with CO₂.

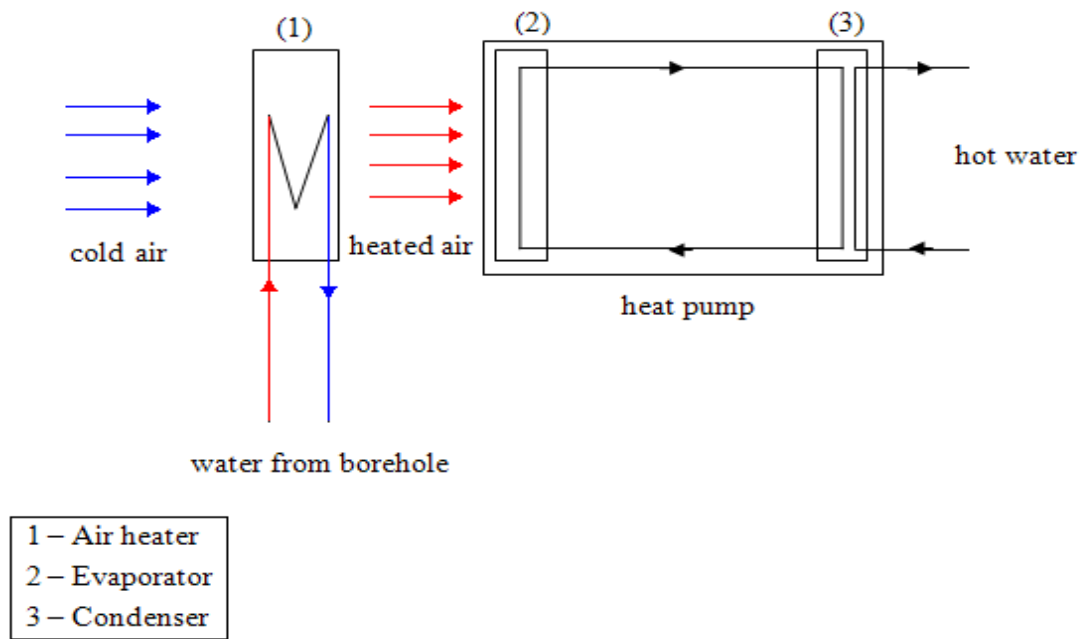


Figure 4 A schematic of the heat pump installed in Grýtubakki, NE-Iceland

2 HEAT PUMP THEORY

2.1 Reasons to use heat pumps

During times of cheap and easily accessible energy no one seriously considered installing a heat pump inside their home – it was only in the domain of scientists and engineers. After the oil crisis in the USA in the 1970' (caused by reaching oil peak in

North America) people realized, maybe for the first time in a large scale, how precious energy is. It was the time when the first generation of heat pumps (as well as many other devices) were introduced. These units were primitive, ineffective and in many cases improperly installed. When the crisis was over and cheap, imported oil satisfied demand in the USA people abandoned heat pumps, but heat pump engineers didn't give up. They were improving technologies and are today prepared for the next energy crisis. Heat pumps are becoming more and more popular. Several reasons are responsible for this process:

1. Need to reduce CO₂ emissions - heat pumps utilize energy contained in the ambient environment (it can be energy contained in the air, ground or water) and the only need for a supply of energy is to drive a compressor. Usually in new heat pump units there needs to be delivered about one portion of energy to receive four. Of course the quality of energy is also important; because we can convert electricity to heat without any losses, but we cannot do the opposite (we are limited to the second law of thermodynamics).

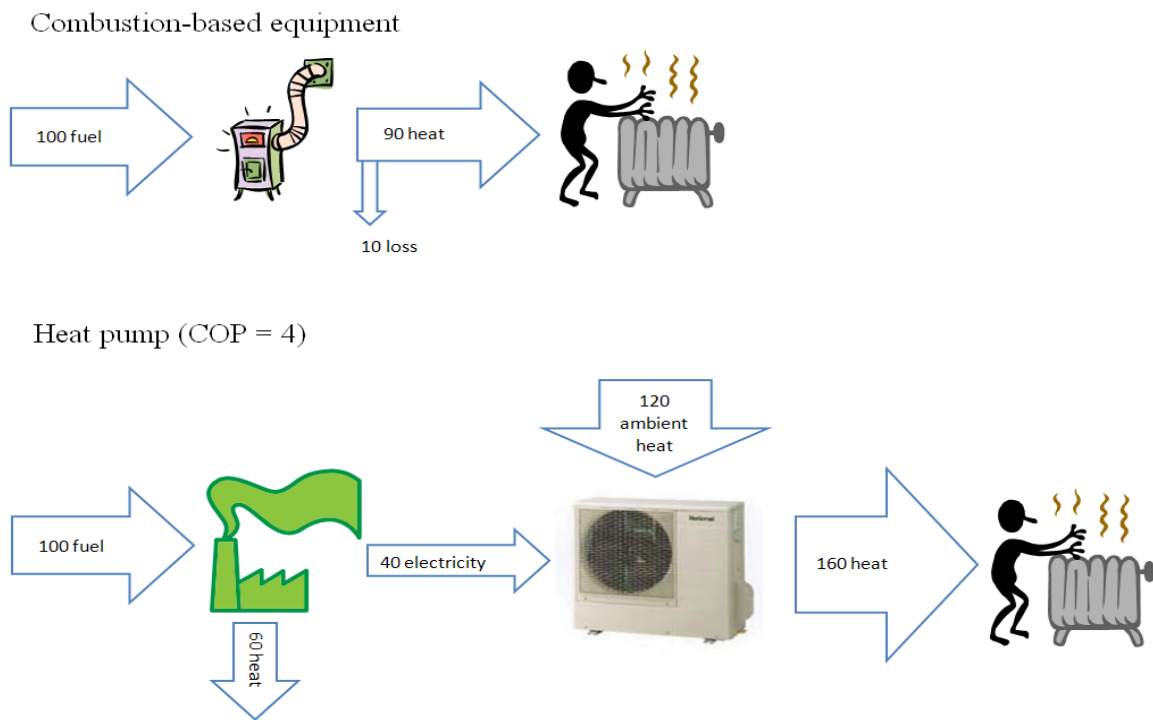


Figure 5 Comparison of energy efficiency between combustion-based equipment and a heat pump

Anyway, if two cases are assumed: combustion based equipment and a heat pump, it will turn out that from 100 units of energy contained in primary fuel only 90 units of heat will be obtained in the first case. By using a heat pump, from 100 units of energy contained in primary fuel 160 units of heat will be obtained. Figure 5 describes this comparison.

This means that from 100 units of energy, 70 more units of heat are generated in the case of the heat pump. In other words, there are savings of 43,75 % of primary energy ($100\% - (90 \times 100\%) / 160$). These calculations are based on three basic assumptions:

- Electricity is made in coal power plant – 40% efficient. This technology is not state of art. Besides that we can produce electricity in a gas turbine coupled with a steam turbine at 55% -60%. Or from renewable or nuclear plants which are, during normal operations, not associated with air pollution.
- Assumed coefficient of performance - COP (see chapter 2.2.1) of the heat pump is 4. Today higher values can be achieved.
- Heat produced in the power plant is rejected to the ambient, but in fact it can be usefully utilized (e.g. in combined heat and power systems).

Assuming combined heat and power - CHP (40% electricity, 40% useful heat), COP = 5 it is possible to see how previous calculations will change. In this case, from 100 units of primary energy, 240 units of heat will be generated, which means 250 more units than from a traditional boiler!

According to the IEA Heat Pump Centre, heat pumps can cut global CO₂ emissions by nearly 8%, which corresponds to 1.8 billion tones of CO₂ per year (HeatPumpCenter, 2006). This is equivalent to the CO₂ emission of 244 GW coal-fired power plants (specific emissions 890 kg CO₂/MWh_{el}, operating time 8 400 h/year).

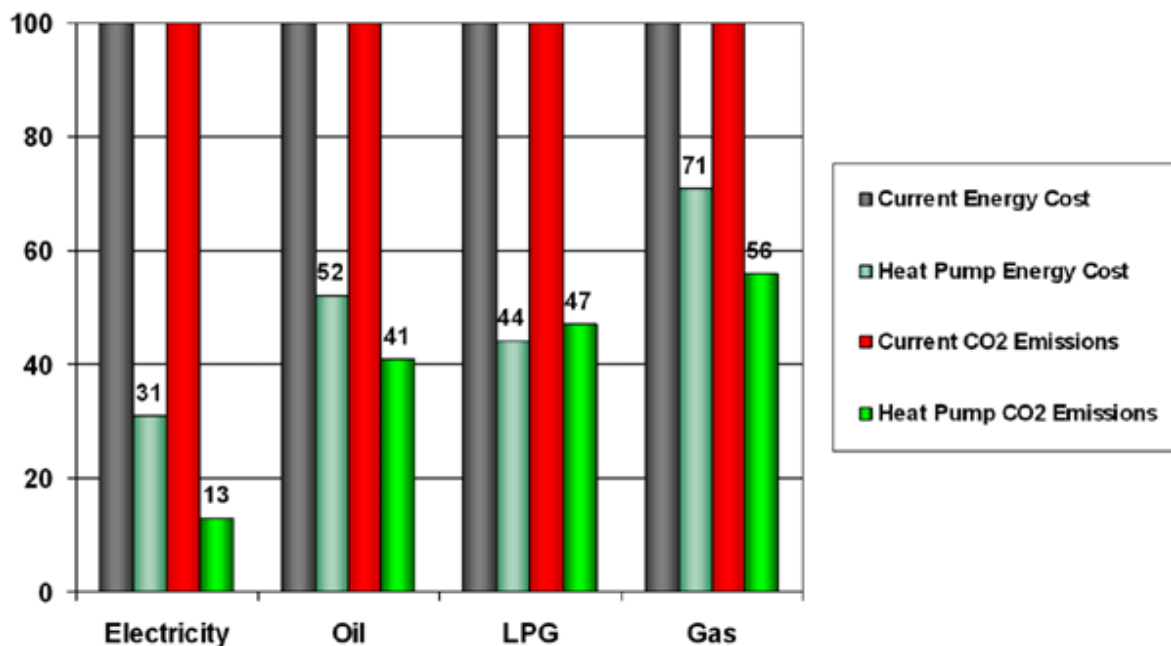


Figure 6 Comparison of costs and CO₂ emissions between heat pumps and commonly used heat sources for space heating (The Fuel Effect, 2008)

2. The second reason, maybe of the same importance, is economy. The initial cost in most residential applications can be two to three times higher than that of a conventional heating system, but operation costs are much lower (figure 6).

3. The third reason is energy security. As it is shown in figure 7, oil reserves are located far from importers and in many cases in countries which are considered unstable, often nondemocratic, and where the state controls the energy sectors. Companies from that sector are driven by politics and are not always working according to market based rules. There is a threat that the fossil fuels they export might be used as a political tool to have an impact on other countries. Of course coal reserves are more abundant and located almost world-wide, but the fossil fuel market is connected, and if the price of one goes up, there is a higher demand on other suppliers for their prices to follow the increasing trend (figure 8). Heat pumps can be installed in all parts of the world. To run them we need electric energy which can be produced from various sources and easily transmitted.

Who has the oil?

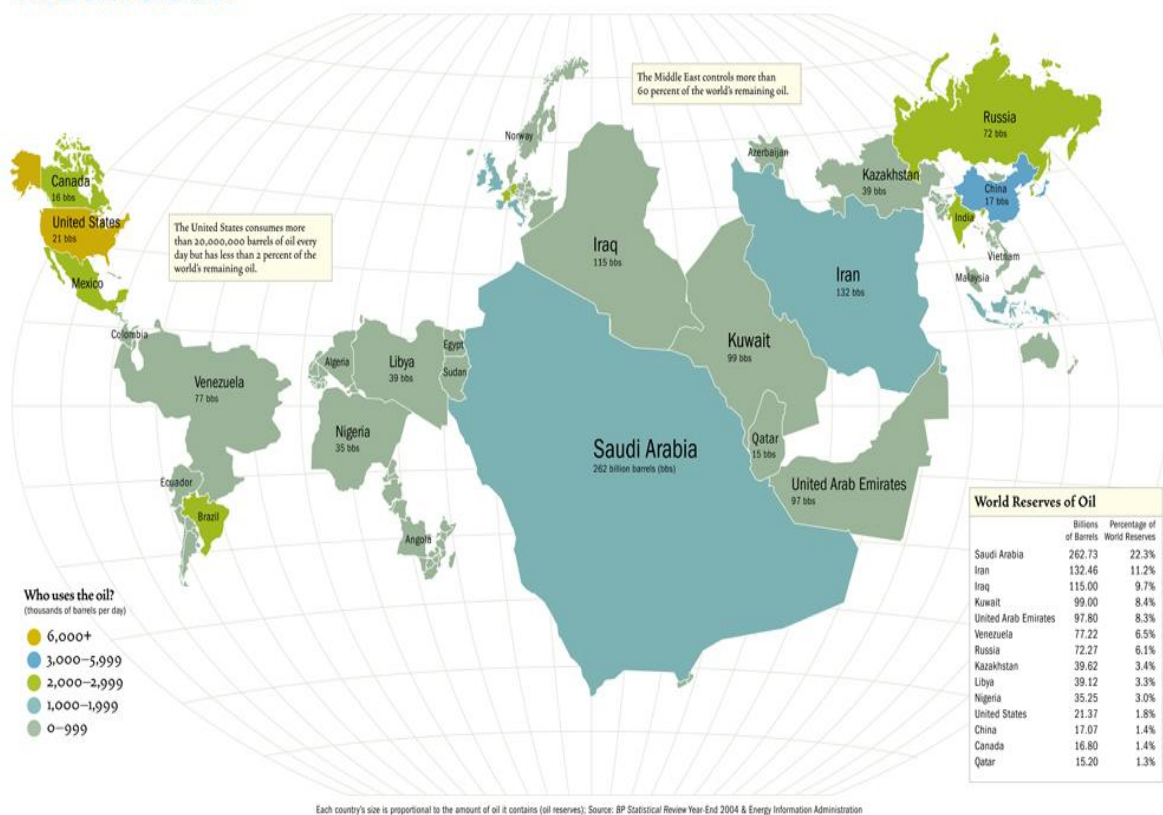


Figure 7 Distribution of the oil resources (BP, 2004)

Energy price trends (1987-2005)

Source: BP 2005

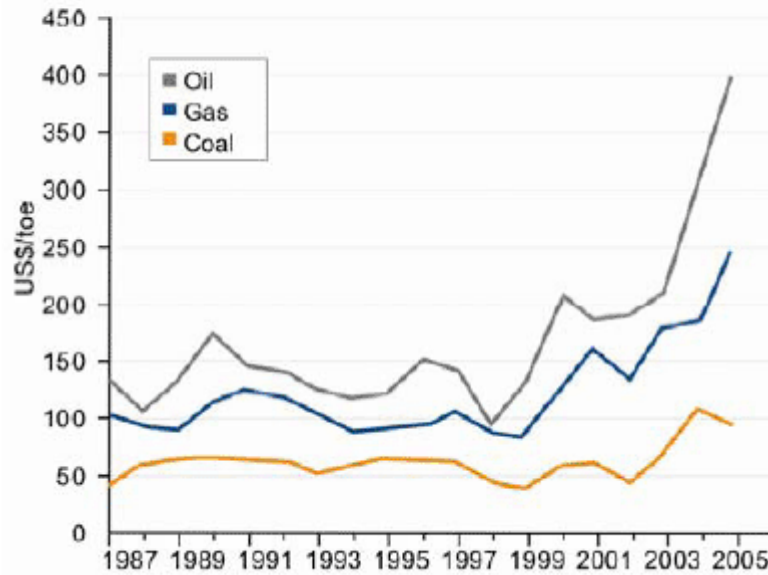


Figure 8 Energy price trends (1987-2005) (WEC, 2008)

4. For the end user, comfort is also significant. Heat pumps are quiet and operate automatically. No combustion takes place and refrigerants are usually nonflammable, so no danger of fire can occur. Heat pumps require only electricity to run. This is important in remote areas where no gas infrastructure is present. Additionally, some heat pumps offer cooling function (passive or active), so no additional device is needed.

2.2 Principles of work

Direction of heat flow in nature is from higher to lower temperature regions. A device which can reverse this process is called a heat pump or refrigerator. It consists of a condenser, a throttling valve, an evaporator and a compressing device (compressor or complex absorption component discussed later). A heat pump takes energy from its surroundings (lower heat source) and by adding work, “pumps” it into a building (upper heat source). This process is shown on figure 8.

2.2.1 COP and other factors

A factor which defines how effective the heat pump is, is called COP – coefficient of performance. It is defined as:

$$COP_{HP} = \frac{\text{Heating effect}}{\text{Work input}} = \frac{\dot{Q}_H}{W_{in}}$$

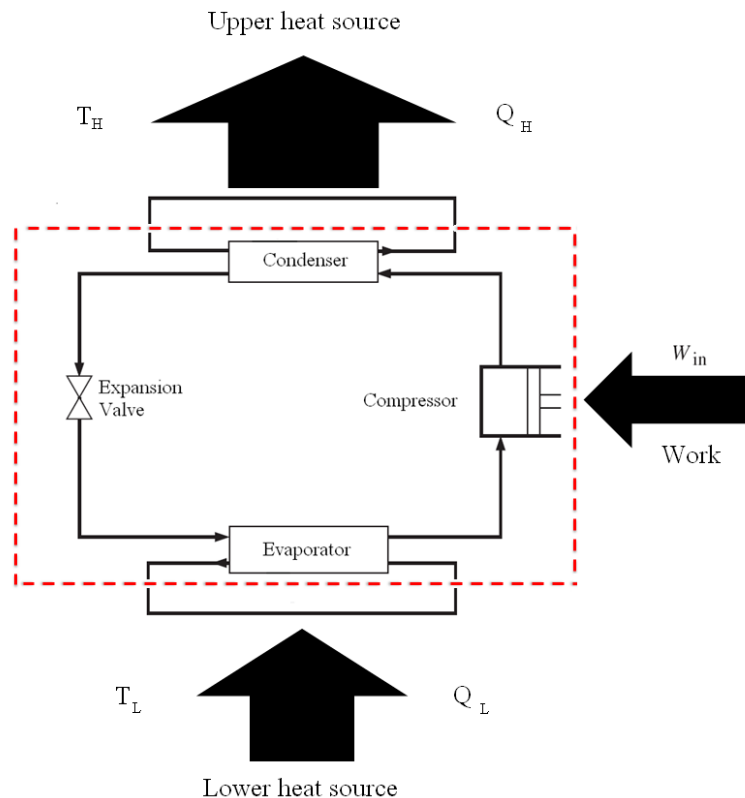


Figure 9 A basic schematic figure of a heat pump

COP depends on heat pump design, properties of the working fluid and temperature difference between the upper and lower heat source. Usually COP is given by a producer under specific input and output conditions – e.g. B0/W35, which simply means: water temperature to an evaporator of 0°C and output water temperature from a condenser of 35°C. In reality it is impossible to keep those specified conditions for all heating seasons and heat pump types, so it is more appropriate to use in calculations the so-called Seasonal Performance Factor, also named Annual Efficiency. It is simply the ratio of average total annual heat energy provided by the heat pump to the average total annual electric energy used over the same period.

2.2.2 Thermodynamic cycles

Heat pumps work in a repetitive way that can be described by ideal cycles. These cycles are often shown on diagrams. The most popular are the T-s (temperature – specific entropy) diagram and the P-h (pressure – specific enthalpy) diagram.

The reversed Carnot cycle

There are two ideal cycles for heat pumps. The first is called the reversed Carnot cycle. It is the simplest and most thermal efficient for given temperature limits, but doesn't describe the actual process too accurately. It is used as a standard to show how a real process differs from the most thermally efficient one. The cycle consists of two isothermal (1-2, 3-4) and two isentropic processes (2-3, 4-1) (figure 10).

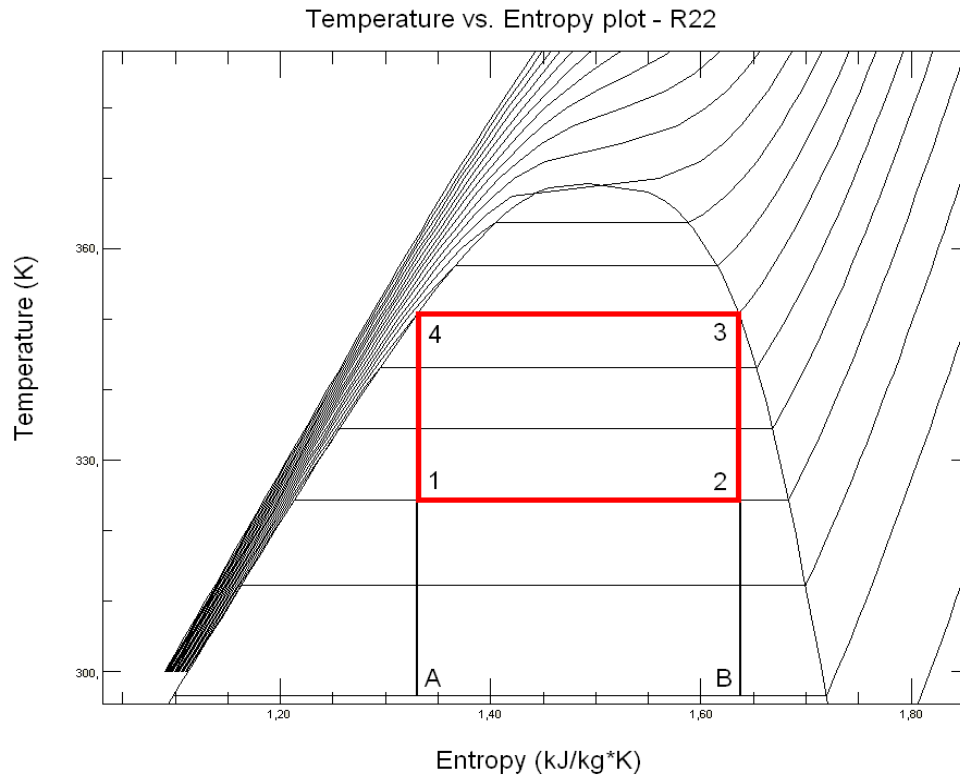


Figure 10 The reversed Carnot cycle presented in T-s diagram for working fluid R-22

COP in reversed Carnot cycle is described by the equation:

$$COP_{HP,Carnot} = \frac{1}{1 - T_L/T_H}$$

Notice that COP rises if temperature difference ($\Delta T = T_H - T_L$) decreases. The relationship between COP and temperature difference is shown in figure 11.

The ideal vapor-compression refrigeration cycle

The second ideal cycle is called the ideal vapor-compression refrigeration cycle. It is the most popular because it accurately describes real processes (figure 12). It is often shown on T-s and P-h diagrams. The cycle consists of four processes:

- 1-2 compression in compressor (constant entropy)
- 2-3 heat rejected in condenser (constant pressure)
- 3-4 throttling in valve (constant enthalpy)
- 4-1 heat absorption in evaporator (constant pressure)

$$COP_{HP} = \frac{h_2 - h_3}{h_2 - h_1}$$

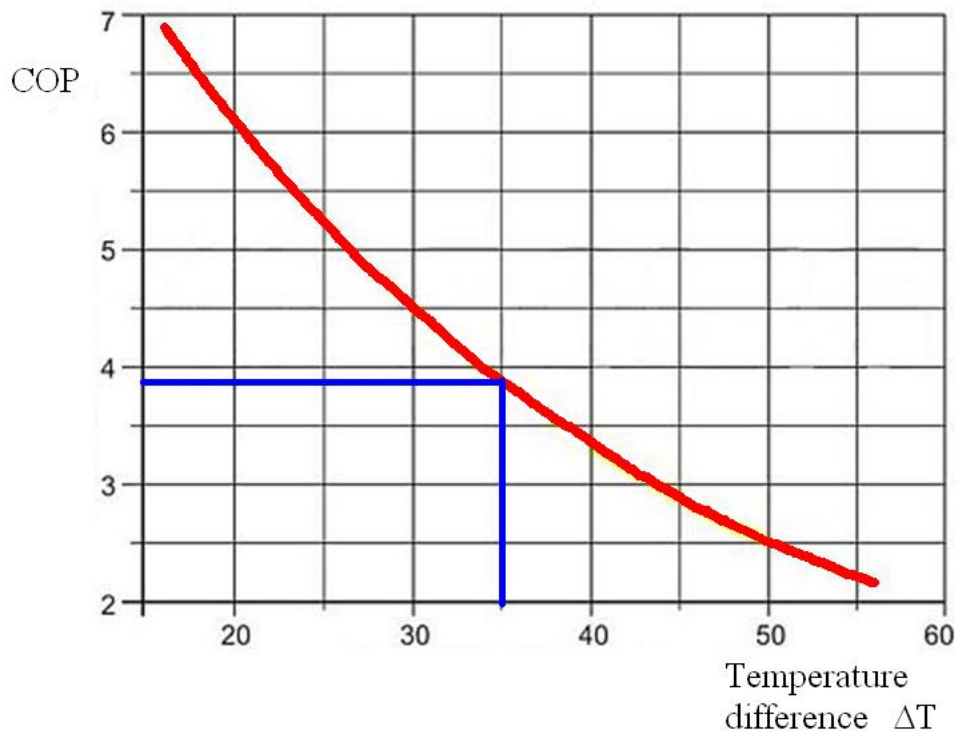


Figure 11 A relationship between COP and temperature difference between a lower heat source and an upper heat source

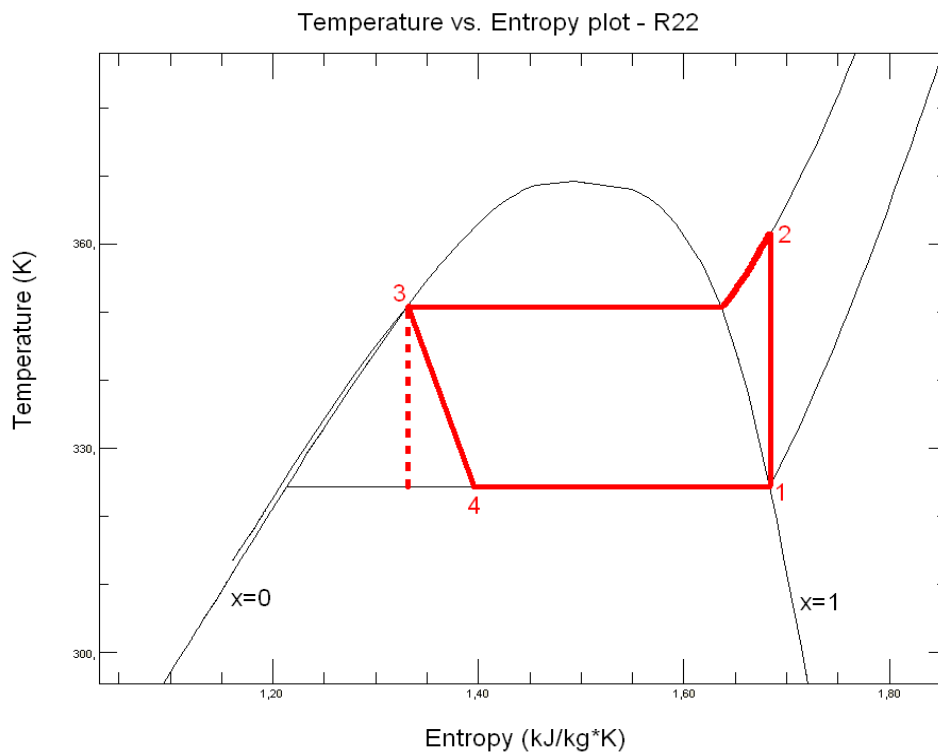


Figure 12 The ideal vapor-compression refrigerant cycle presented in T-s diagram for the working fluid R-22

The actual cycle

An actual process is more properly presented in figure 13. In ideal cycles, no entropy is generated and processes can be described as reversible. In reality, all processes are irreversible, mostly because of pressure drop (fluid friction) or heat losses to the surrounding.

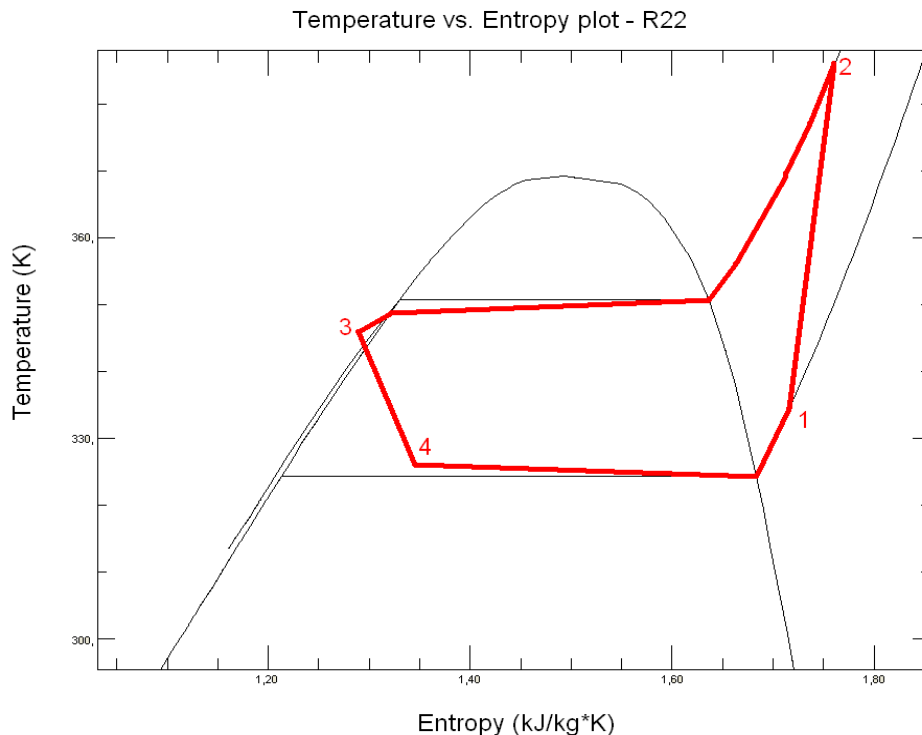


Figure 13 An approximation of the actual process presented in T-s diagram for working fluid R-22

2.2.3 Absorption heat pumps

Another category of heat pumps are absorption heat pumps. Generally, in such devices the compressor component is replaced by the absorption component (figure 14). In the absorption process two types of fluids interact with each other: the first is called the refrigerant (it circulates between absorption part, condenser, throttling valve and evaporator) and transports medium, which only circulates in the absorption component. The absorption segment consists of an absorber, generator, pump, rectifier, regenerator and expansion valve. When the refrigerant leaves the evaporator it enters an absorber where the absorption reaction takes place. It is an exothermic reaction, so some amount of heat is released, which is usually separated in the cooling tower. The solubility of the refrigerant in the transport medium depends on the temperature in the absorber. The more heat taken off from the absorber, the more refrigerant will dissolve. Mixed fluid leaves the absorber and enters the pump to the regenerator where some amount of heat is added to the mixture (heat added from an external heat source – waste heat, solar or geothermal energy) and

some part of the mixture evaporates. The evaporated mixture is rich in refrigerant and it goes to the rectifier where the rests of the transport medium is separated and returned back to the generator. Pure high temperature and high pressure refrigerant goes to the condenser. The part of the mixture in the generator which did not evaporate (weak in refrigerant) is returned to the absorber. On its way there it rejects heat in the regenerator.

Absorption devices are less efficient, more expensive, occupy more space, and are more complex than compressors, so they should be considered only in locations where cheap thermal energy is available (the amount of electricity needed to drive this device is negligible). In households they are mainly gas-fired, but in industry they are driven by waste heat or high pressure steam.

The most common working fluids used in absorption systems are:

- Water (refrigerant) and lithium bromide (transport medium)
- Ammonia (refrigerant) and water (transport medium)

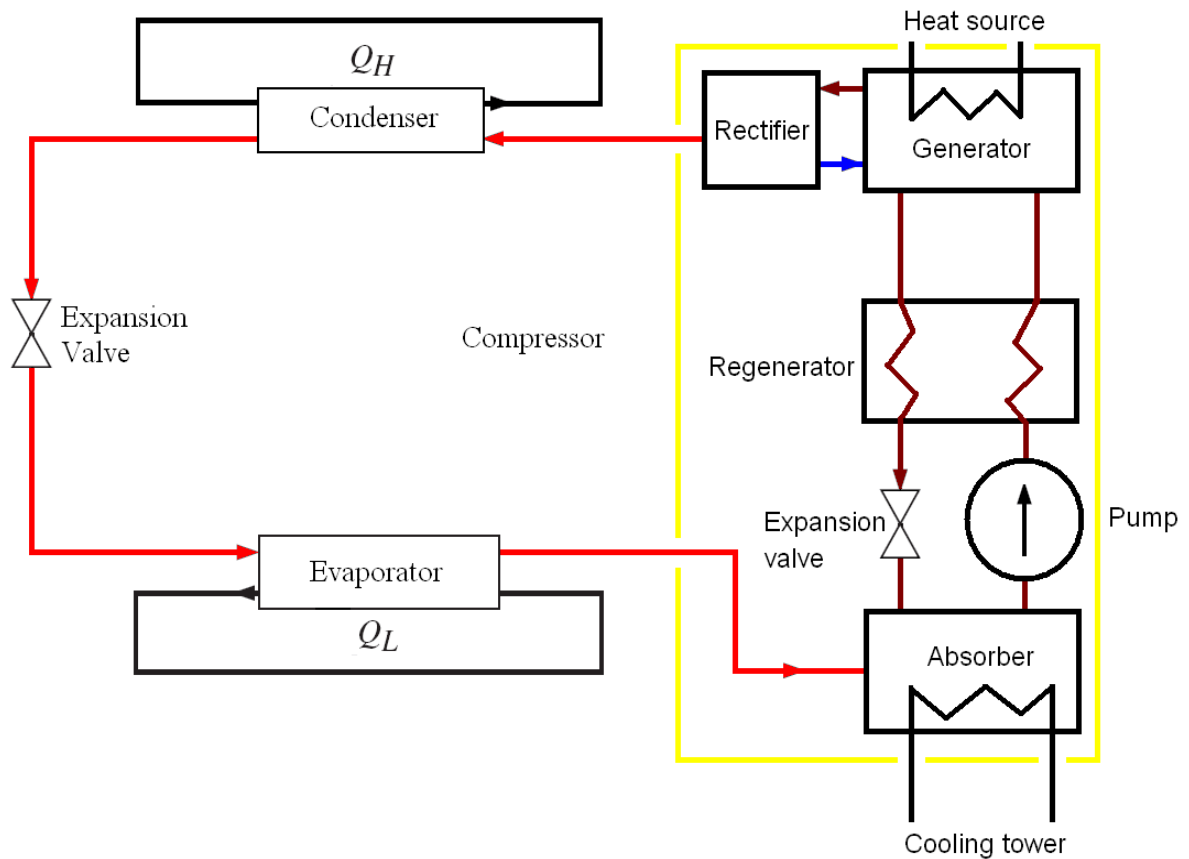


Figure 14 A basic schematic of an absorption heat pump

2.2.4 Refrigerants

The history of heat pumps begins with the discovery of the Carnot cycle in 1824. This discovery provided the theoretical background and showed that it is possible to construct a device which will cool things down below the temperature of the surroundings. Later, when continued research lead to the development of new theoretical tools and the first machines, the evolution of refrigerants started. A few trials were made with different

substances, until in 1930 Thomas Midgley discovered dichlorodifluoromethane, a chlorinated fluorocarbon known as Freon (chlorofluorocarbons - CFC). It seemed to be a perfect solution until in the 1980s when people realized that it is harmful for the ozone layer. CFCs were gradually replaced by hydrochlorofluorocarbons - HCFCs (which had smaller ozone depletion potential - ODP) and next with HFC (hydrofluorocarbons) with ODP equal to zero. Unfortunately they all have relatively high GWP - global warming potential (which is a measure of how much a given mass of greenhouse gas is estimated to contribute to global warming).

When people realized that global warming was caused by them, scientist started research on natural refrigerants with ODP and GWP zero or near-zero. It is clear that refrigerants contained in heat pumps make a small contribution to global warming (compared to emissions from power plants or internal combustion engines), but the heat pump market is growing and the same working fluids are used in air-conditioners, so the choice between refrigerants is important. It turned out that among natural working fluids for wider use, CO₂ is the best choice since ammonia and hydrocarbons are toxic or flammable (and should be used more carefully), and water is excellent but only for high temperature industrial use.

In the past, the choice of working fluid for a heat pump was much easier, because the crucial criterion were physical properties. Now it is a complicated issue: it has to be a compromise between the best physical and chemical properties on the one hand, and environmental protection on the other. Physical properties mean thermal conductivity, critical point, heat of evaporation; chemical refer to toxicity, corrosive properties and by environmental protection – ODP and GWP. There are five commonly used types of working fluids: CFCs, HCFCs, HFCs, blends and natural fluids. Classification of commonly used working fluids is presented in table 1.

Table 1 Classification of commonly used working fluids

Type of working fluid	Examples	ODP	GWP	Remarks
CFCs	R-11, R-12, R-13, R-114, R-115, R-500, R-502, R-13B1	High		Prohibited refrigerants. Phased out 1 January 1996 (IEAHeatPumpCentre, 2008).
HCFCs	R-22, R-401, R-403, R-408, R-409	Medium		Called transitional refrigerants, also contain chlorine. To be phased out by 1 January 2020 (IEAHeatPumpCentre, 2008).
HFCs	R-134a, R-152a, R-32, R-125, R-507	None	Medium	Chlorine free – can be considered as an alternative for CFCs and HCFCs.
Blends	R-407-C, R-410-A	None	Medium	Mixtures of 2 or more pure refrigerants can be well adjusted for particular conditions and needs.
Natural	Water, ammonia, carbon dioxide, hydrocarbons	Zero or near-zero		Water – excellent for high temperature industrial heat pumps. CO ₂ – discussed later. Hydrocarbons – flammable. Ammonia – excellent thermodynamic properties, but toxic and flammable.

2.3 Lower heat source

Heat pumps take energy from their surroundings, which will be called LHS (lower heat source). The more heat is taken from the environment the less work needs to be added to receive the same output, so the choice and calculations of LHS are most important during the process of heat pump system planning. A perfect heat source should have high and stable temperature, good physical and chemical properties.

Generally there are two types of LHS:

- Open systems (figure 15)
- Close loops (figure 16)

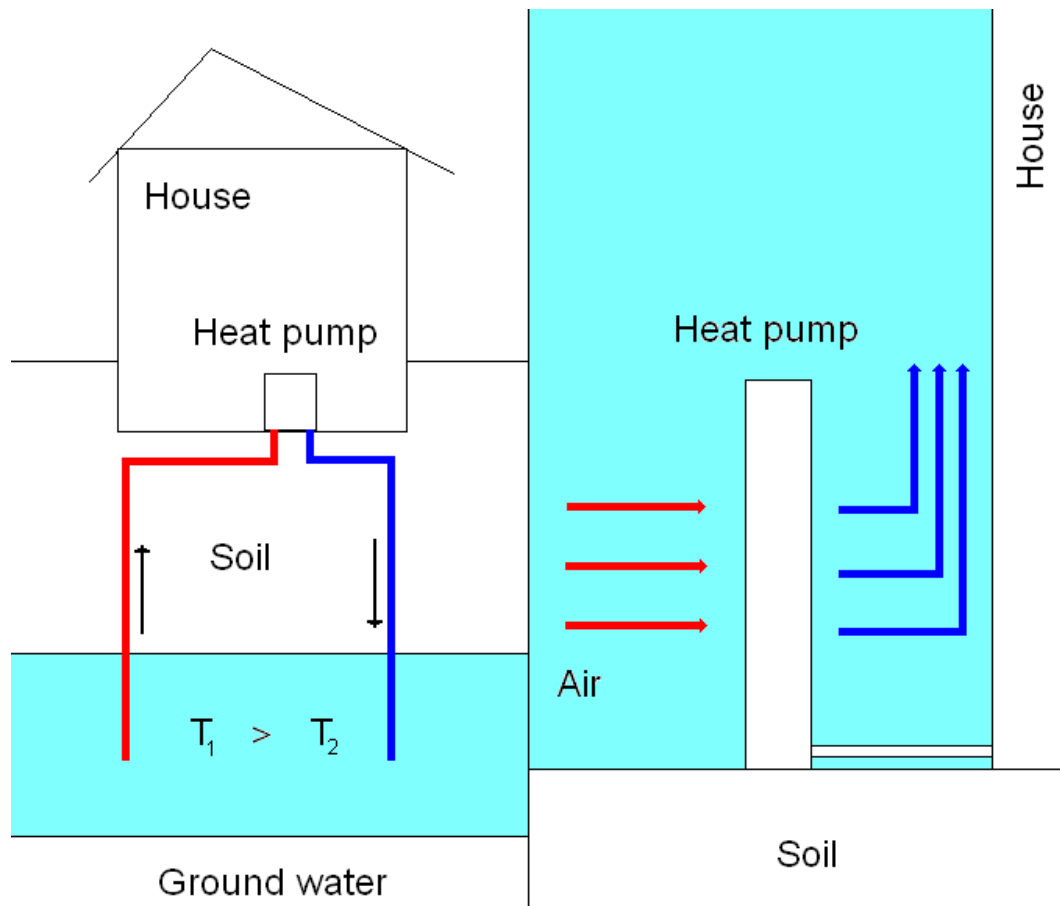


Figure 15 Types of the most often installed open systems

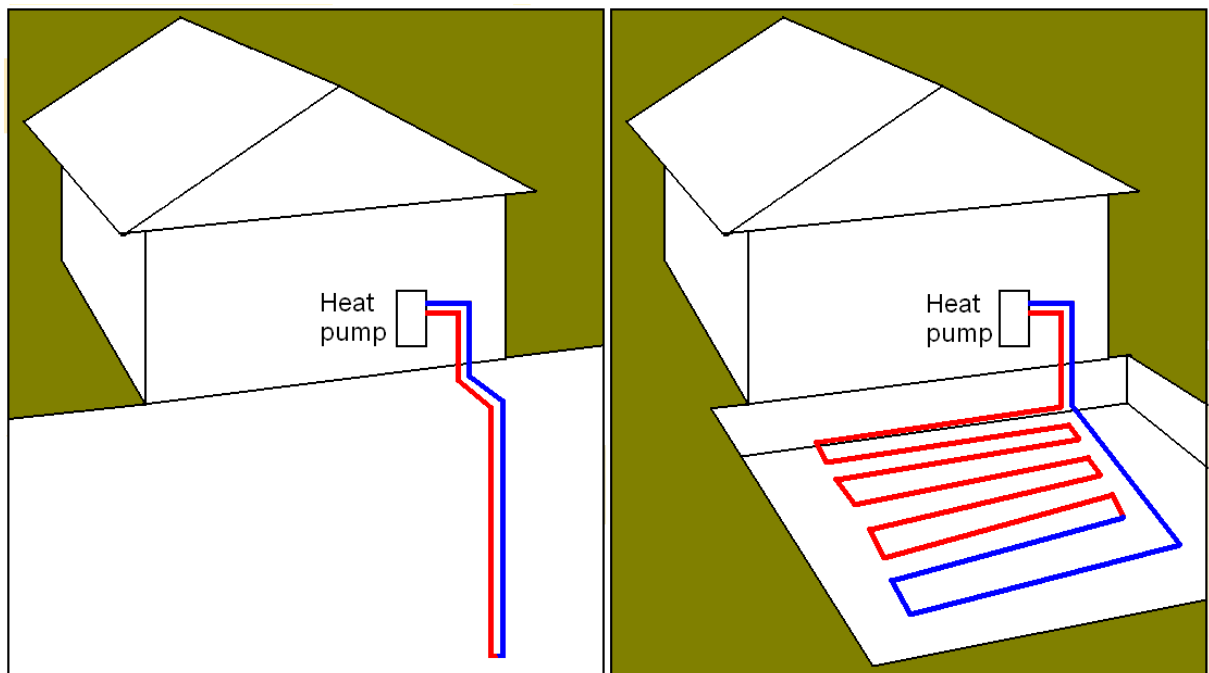


Figure 16 Types of the most often installed close loops systems

In this first case fluid is taken from the surrounding and after it transfers its energy to a heat pump, it is returned back. To this group belongs: ambient air, exhaust air, ground water. For larger systems, usually industrial systems, waste water, sea, lake and river water can be used.

Closed loops are more popular. In this type LHS is connected to a heat pump by a system of tubes in which non-freezing fluid circulates.

A comparison of the most often used LHSs is shown in table 2.

Table 2 The most often used lower heat sources – temperatures and properties

LHS	Temperature range [°C]	Remarks
Ambient air	-10, 15	Low investment costs, no impact on the ground, lower seasonal performance by 10%-30% than water LHS, defrosting function needed, no passive cooling allowed during summer
Exhaust air (ventilation)	15, 25	Continuous operation of ventilation needed during heating period, no passive cooling allowed during summer. Low investment costs.
Ground water	4, 10	High initial costs, small area needed, stable temperature, passive cooling possible, drilling needed, local law regulations.
Lake, river water	0, 10	Lake or river needed near the building, risk of freezing, relatively small costs.
Ground – vertical heat exchanger	5, 10	Large investments costs, stable temperature during whole year, passive cooling allowed, drilling needed, local law regulations
Ground - horizontal heat exchanger	0, 5	Large area needed, relatively high costs.
Waste water	>10	Relatively high and constant temperature throughout the year, constant waste water flow needed.

2.4 Upper heat source

It is important while planning the installment of a heat pump to choose one of proper power. If it is too small there will not be sufficient power to supply enough energy to the building and reach the desired temperatures. If it is too large, costs will rise and it will not work at optimal conditions. The simplest way to estimate heat pump power for existing buildings is to check the cost of energy used to warm up the building in previous heating seasons. More complicated are the cases when there are no heating bills, it is necessary to modify a building or deal with a new one. Then a commercial program can be used, such as HVAC-Calc 4.0 or heat source power can be calculated. In appendix B assumptions and techniques needed to estimate heat demand for a small buildings are attached.

3 CO₂ HEAT PUMPS

In the following chapter the EcoCute CO₂ heat pump will be presented with possible changes which could improve its functionality when used with low temperature geothermal sources.

The EcoCute heat pump was developed in 2001 in collaboration between TEPCO, Denso and CRIEPI. It is commercially available on the market as an air-to-water unit (see figure 17). It attracts attention because it is an environmental friendly product (CO₂ as a working fluid) with the ability to work effectively at a wide range of temperatures. Since EcoCute was developed in 2001 its market has grown rapidly. In Japan it is assumed to reach 5.2 million units installed in 2010, with over 1 million units sold already. Its export is also increasing significantly (figure 18).



Figure 17 Commercially available EcoCute models made by different companies (Hashimoto, 2006)

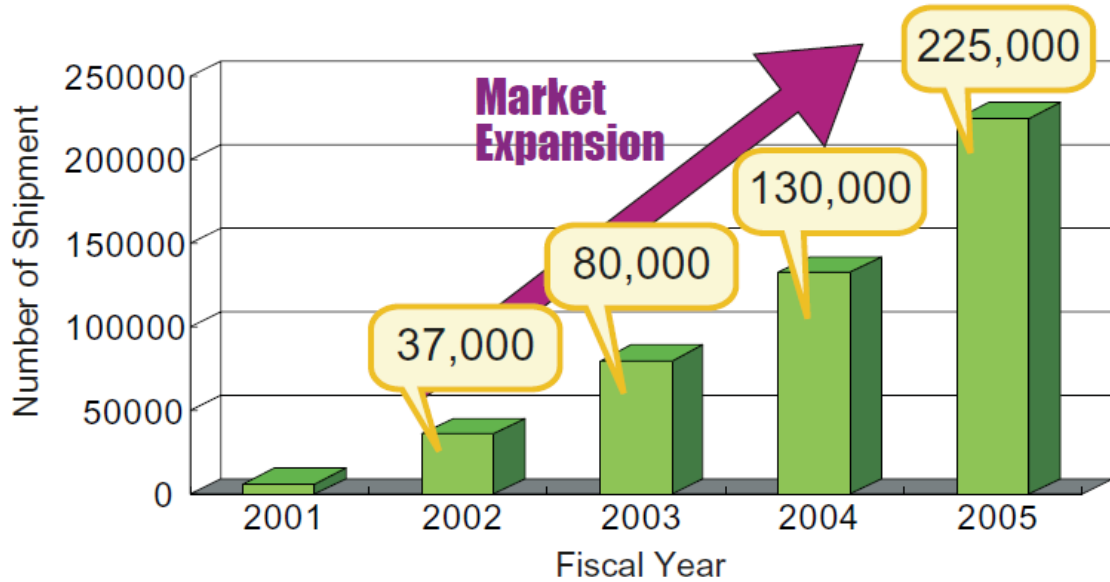


Figure 18 Shipment of EcoCute units (Hashimoto, 2006)

3.1 Reasons to use

There are several reasons to use CO₂ heat pumps in Iceland:

1. 9% of Icelandic houses are heated with electricity (Samorka, 2009), mostly at the Western Fjords and east of the country. Prices of electric energy are heavily subsidized by the state. Assuming average seasonal COP to be 4, 75% of electricity can be saved. This means that the government could stop the majority of subsidies without affecting the householder's standard of living.

2. Environmentally friendly

Group	Type	ODP	GWP
Natural refrigerant	CO ₂	0	1
HFC	R-410A	0	1900
HFC	R-407C	0	1600
HCFC	R-22	0.055	1700

3. Low ambient operation temperature (as air-to water unit)

Type of refrigerant	Approximately the lowest ambient temperature that the refrigerant can operate at
HFC or HCFC	-10
CO ₂	-20

4. Not toxic and nonflammable.

5. High temperature rises and high COP can be expected in the heating process

3.2 Construction

The CO₂ heat pump consists of a compressor, evaporator, condenser and throttling valve. All these elements were changed mainly due to high pressure and properties of the working fluid. Below, a few important parts will be presented; except the evaporator, which will be described carefully in chapter 4.

3.2.1 Compressor

The heart of EcoCute heat pump is a 2-stage rotary compressor. The unit was made in the year 2000 by Sanyo Electric for air-to-water heat pump. In the first step it compresses carbon dioxide from low pressure (1.4-3.9 MPa) to intermediate (6-8 MPa), in the second step it reaches 11-12.5 MPa (figure 19).

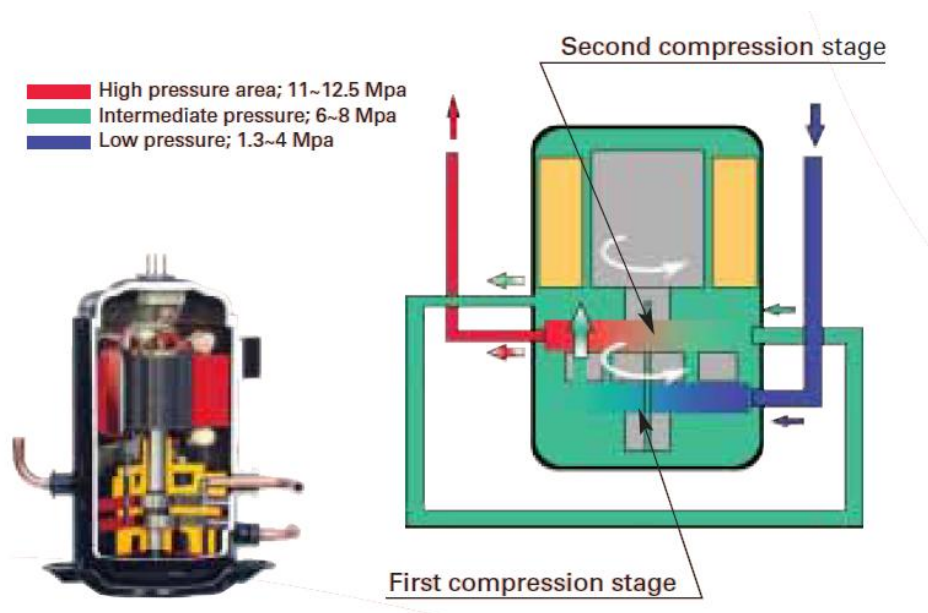


Figure 19 A cutaway of two-stage rotary compressor (SANYO, 2006)

On the T-s diagram compression starts at point B and ends at point C (figure 20). To get high COP the compression process needs start at the point of highest enthalpy, i.e. at the highest temperature and pressure. Since pressure is limited to 3.9 MPa at the inlet to the compressor, it means that the inlet temperature should be increased so that it is as close as possible to the temperature of the LHS. If the LHS is air, this is not always possible because the ambient temperature strongly varies during the heating period.

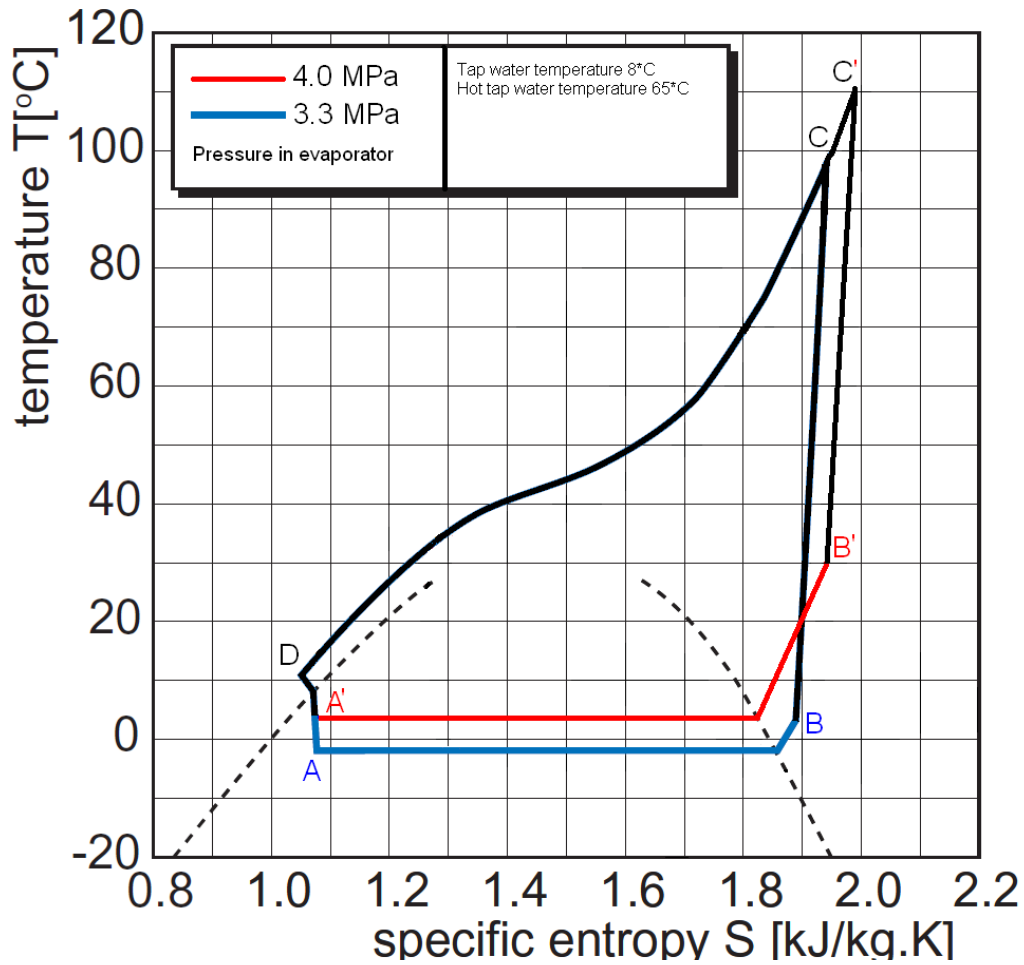


Figure 20 An approximation of actual cycle for EcoCute heat pump presented on T-s diagram.

Heat pumps are constructed in such a way that the pressure at the inlet to the compressor (B) is almost the same (pressure losses due to fluid friction) as at the inlet to the evaporator (A). The process of evaporation occurs in the two phase region, where temperature depends on pressure. For 3.9 MPa, the corresponding temperature is equal to 4.3°C (table 3), so in order to force CO₂ to evaporate, the air temperature should be above this value. To solve this problem the compressor in EcoCute was designed to be operational at variable speed. If ambient temperature drops, then CO₂ expands to lower pressure. For example: the lowest ambient temperature at which the heat pump can operate is -20°C (SANYO, 2006), so the CO₂ evaporation temperature should be lower than this surrounding temperature. At the lowest possible ambient temperature, pressure at the inlet to the compressor and the evaporator is equal to 1.3 MPa (SANYO, 2006). The corresponding temperature to this pressure in the two phase region is -32.8°C (table 4). With such a low temperature of the working fluid at point 1, there needs to be more work delivered by the compressor to obtain the same heat output.

Table 3 Properties of carbon dioxide for given pressure 3.9 MPa and quality 4%

Temperature (°C)	Pressure (MPa)	Volume (m ³ /kg)	Enthalpy (kJ/kg)	Entropy (kJ/kg-K)	Quality (kg/kg)
---------------------	-------------------	--------------------------------	---------------------	----------------------	--------------------

4,3	3,9	0,0014	220	1,07	0,043
-----	-----	--------	-----	------	-------

Table 4 Saturation temperature for given pressure 1.3 MPa

Temperature (°C)	Pressure (MPa)	Volume (m ³ /kg)	Enthalpy (kJ/kg)	Entropy (kJ/kg-K)	Quality (kg/kg)
-32,8	1,3	0,0021	140,81	0,78	0,043

With low-enthalpy water as the LHS we can omit this problem, because the heat pump has a stable temperature of the LHS during the whole year. This temperature is assumed to be higher than 15°C. It is more than enough to evaporate working fluid with a temperature of 4.3°C, which means that at the inlet to the compressor we will always have 4 MPa.

Possible improvements of compressor

The EcoCute was made for the Japanese market and it doesn't fit "Nordic" weather conditions and living standards, mostly because its heat output is too small – 4.5kW. It would be recommended to increase the power of the compressor to obtain higher heat output and decrease power during periods when eclectic heaters are on.

The CO₂ heat pump with low-temperature water as the LHS would be much more efficient (higher COP) if pressure at point 1 was higher. Now it is limited to 3.9 MPa. Higher pressure means higher enthalpy h_B . COP is described as:

$$COP_{HP} = \frac{h_C - h_D}{h_C - h_A}$$

Higher h_B means a lower difference Δh in the denominator. If the numerator is kept constant then we receive higher COP.

3.2.2 Condenser

The condenser was designed from copper capillary tubes (inside diameter $ID=0.5$ mm, outside diameter $OD=1$ mm). It is shown in figure 21. It was projected to keep the smallest possible temperature difference between water and carbon dioxide.

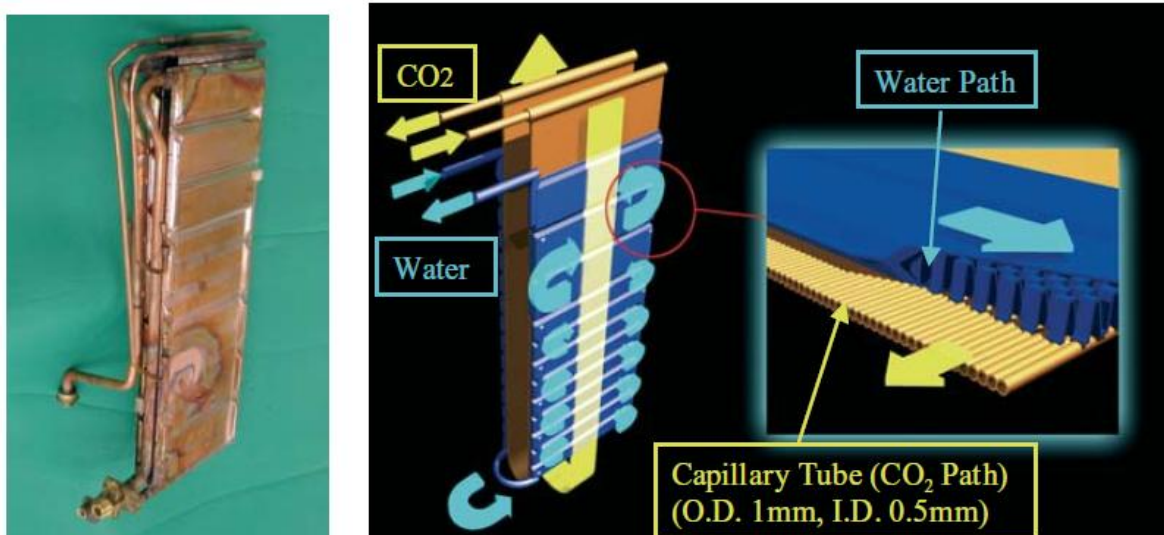


Figure 21 The condenser installed in the EcoCute heat pump (Hashimoto, 2006)

3.2.3 Casing dimensions

All dimensions are presented in figure 22. The original device consists of a heat pump, which is supposed to be installed outside a house, and a storage tank installed inside a building. After the change of LHS it would be recommended to connect them into one unit, which would be installed inside the house. This change would result in higher reliability, because inside the building it could be safe from problems connected with freezing and from mechanical damages. All dimensions of a new casing are shown in figure 23. This design would contain all parts (including the new evaporator) and an unchanged storage tank.

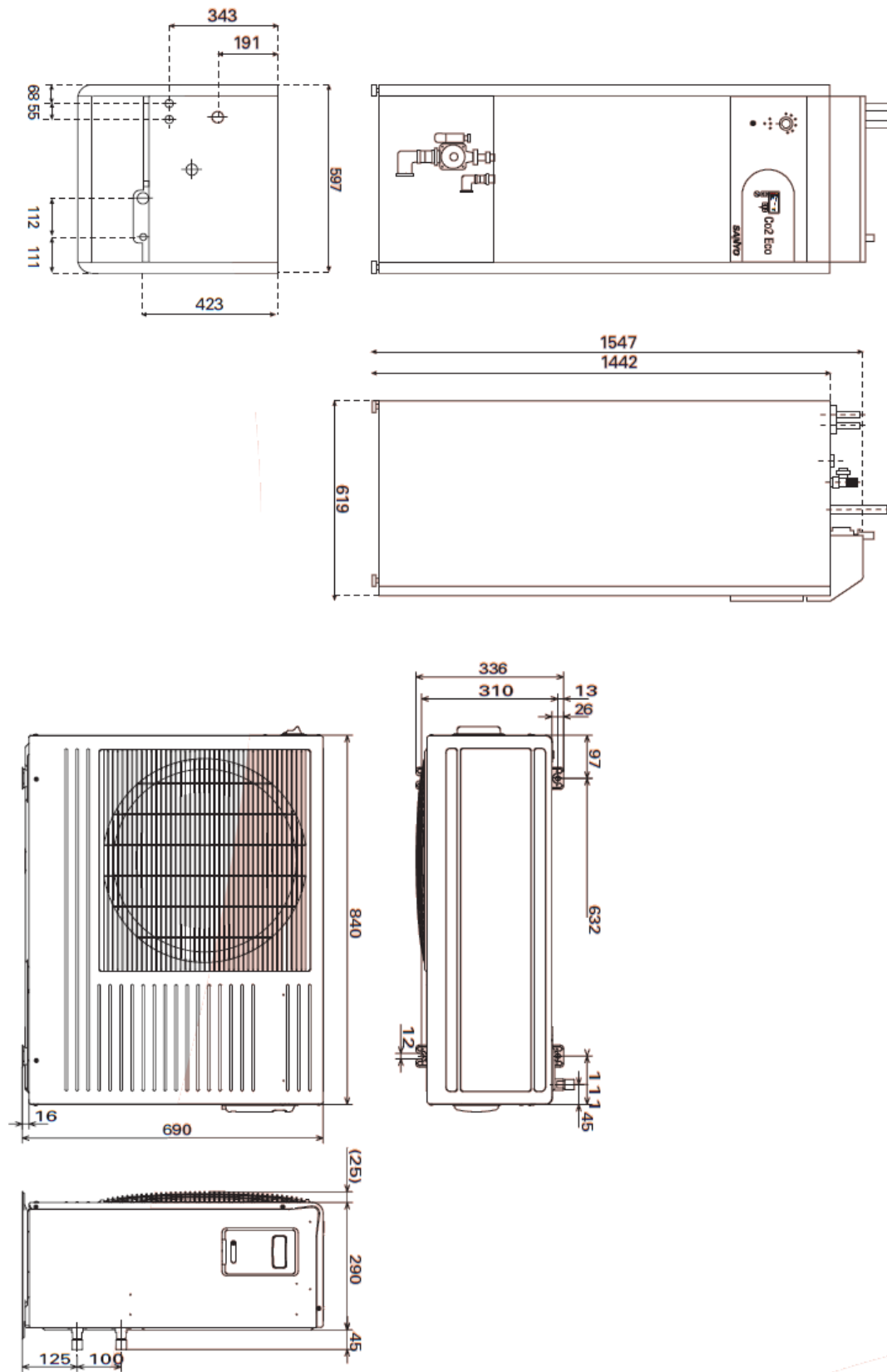


Figure 22 Dimensions of Sanyo EcoCute heat pump. Model SHP-C45DEN (SANYO, 2006)

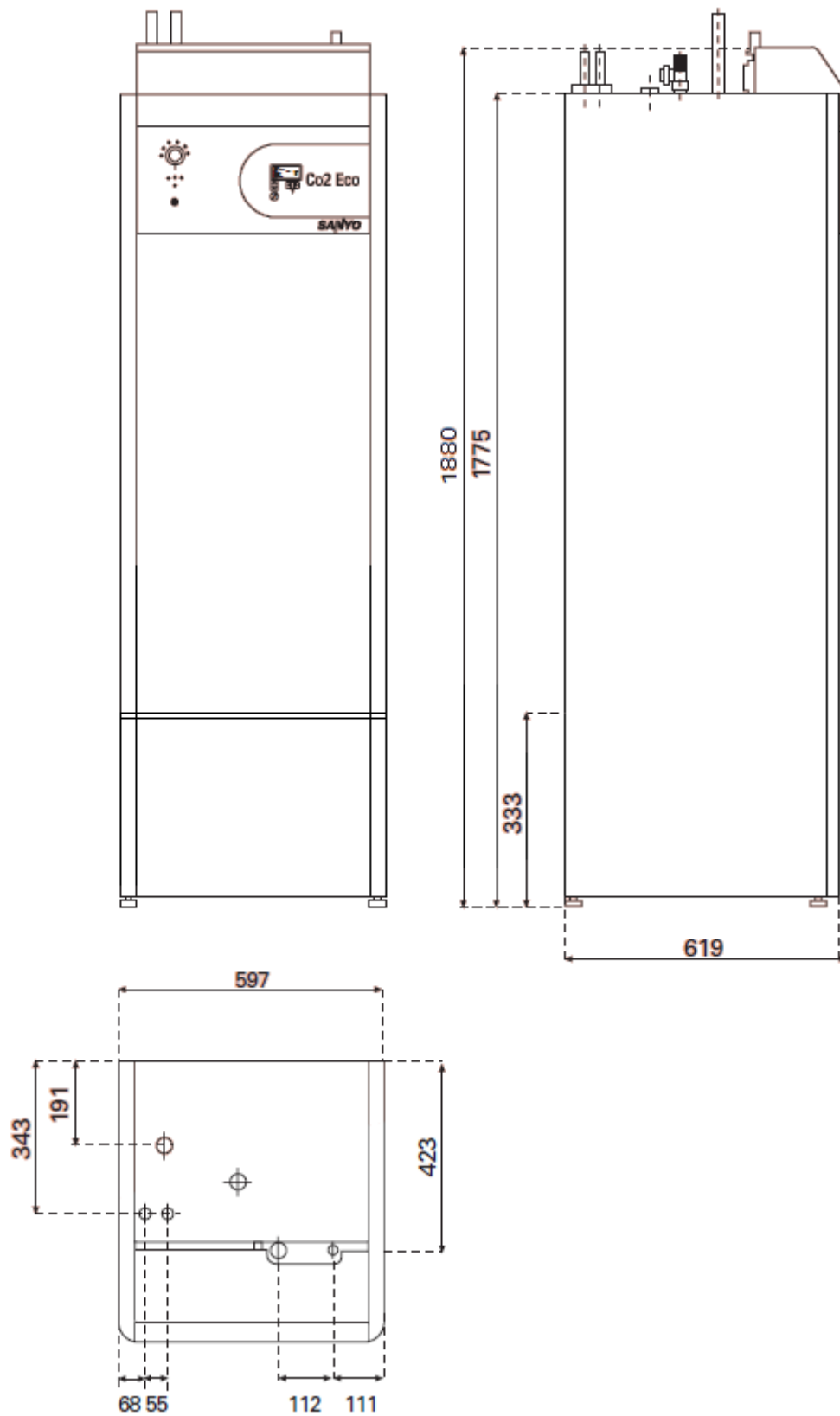


Figure 23 New casing dimensions

4 DESIGN OF THE EVAPORATOR

4.1 Why is the switch of lower heat source from air to water so important?

1. Economy is the most important reason to use water as a heat source. In the air- to- water heat pump COP strongly varies with the ambient temperature, while the water- to- water unit is stable. As shown in figure 24 we obtain COP = 3.75 for the outdoor temperature, equal to 20°C (which is not a typical temperature during the heating season). The average temperature is closer to 0°C, and then COP is about 2.7. When ambient temperature drops and the heat pump is unable to deliver enough energy, electric heaters start working with COP=1. They simply convert high quality electric energy to low quality thermal energy.

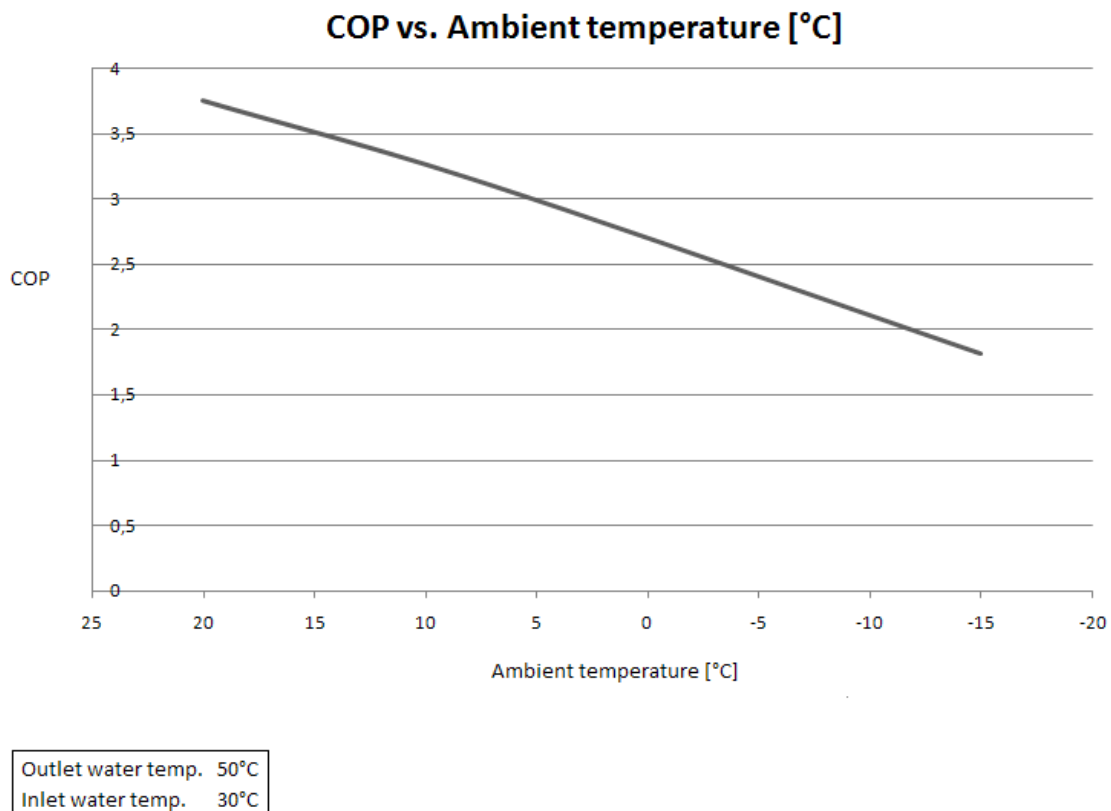


Figure 24 A relationship between COP and ambient temperature of the air to water EcoCute unit

This means that in the case of air as a LHS, COP is not a good factor since it is given for fixed ambient temperature (which in fact strongly varies during the heating season). A better factor (which describes how efficient a heat pump is) is SPF (chapter 2.2.1). In measurements done here in Iceland on the CO₂ heat pump, ISOR proved that SPF rose after installing a heat exchanger in front of the evaporator (figure 25). The heat exchanger was installed on October 16th. Figure 26 shows how power taken by electric heaters was affected by the installation of the heat exchanger. Between the end of June and mid-August no measurements were taken and the

calculated average power over that period is displayed. When the LHS will be changed for water the SPF will grow even higher, which means that more energy will be taken from the ambient and less work done by a compressor. It leads directly to energy savings and lower operating costs.

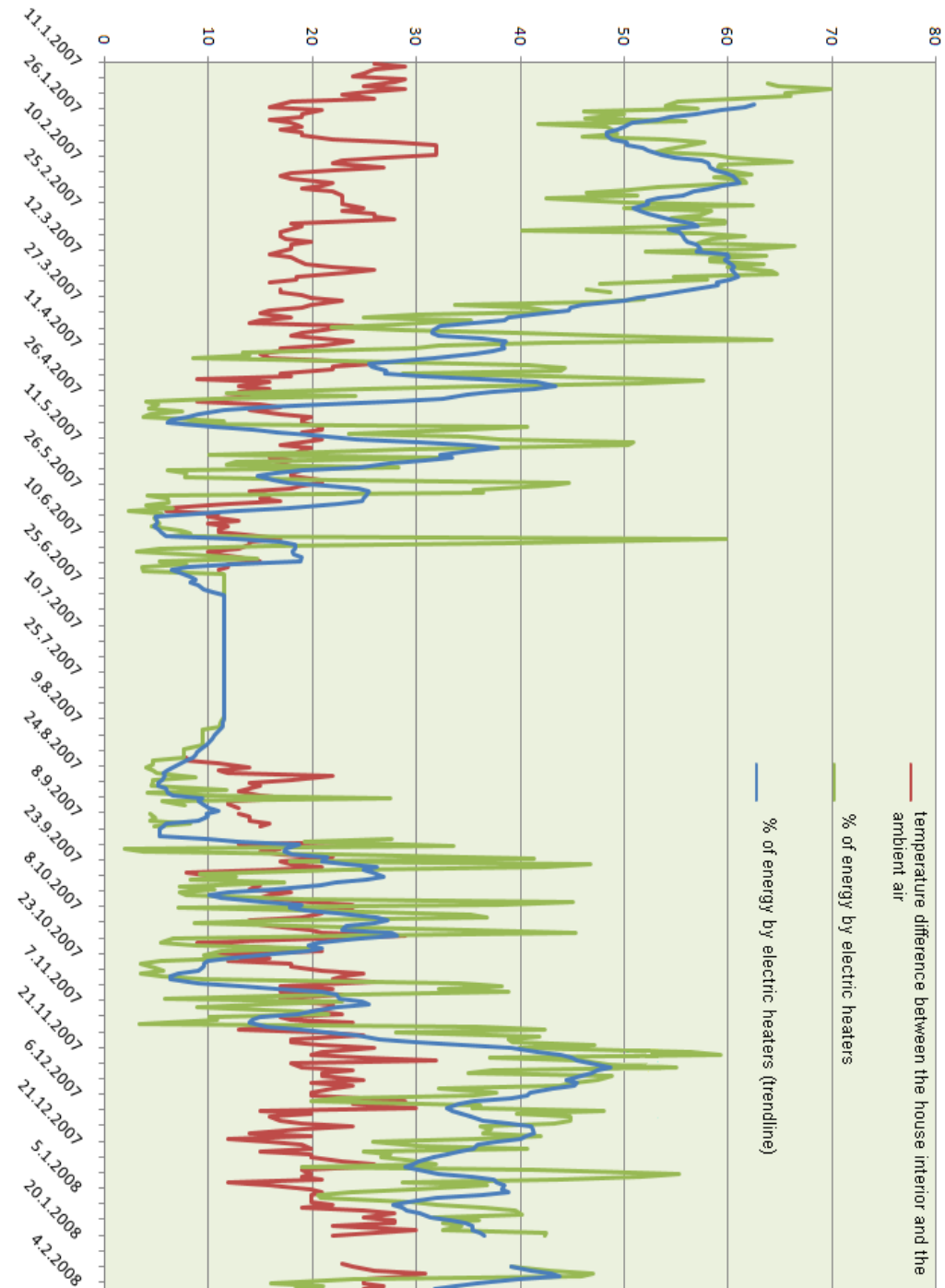


Figure 25 Chart showing measurements done in Grýtubakki, before and after the installation of the heat exchanger in front of the evaporator

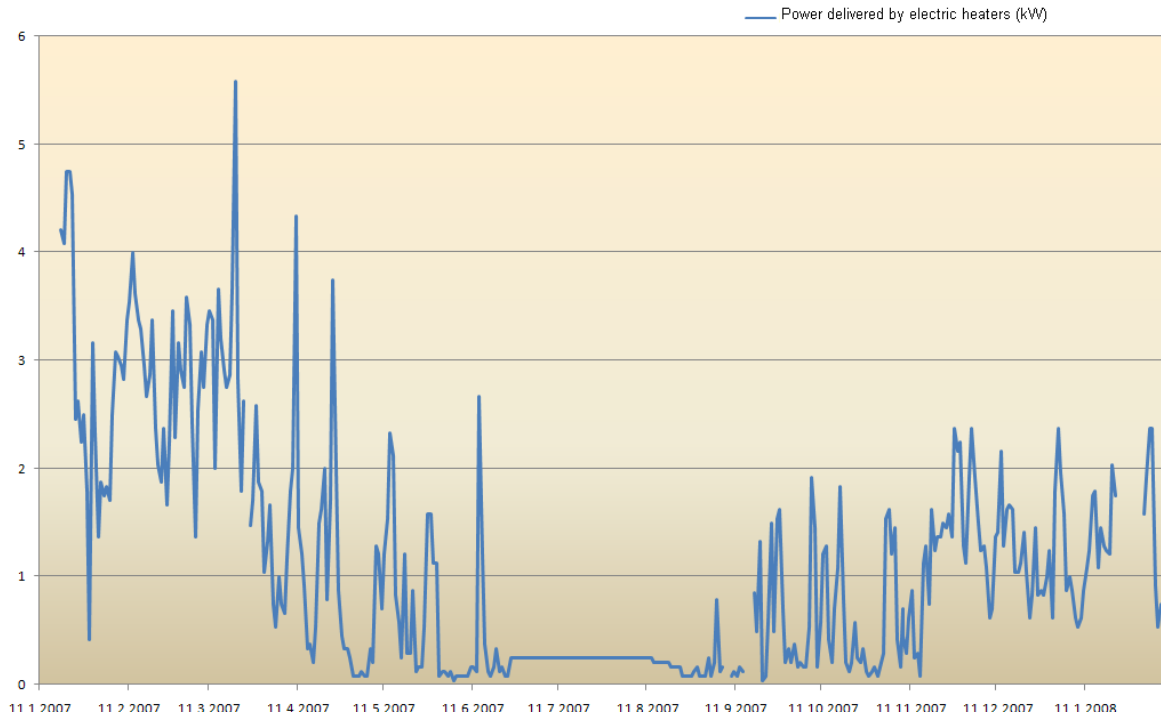


Figure 26 Chart showing power consumed by electric heaters. It decreased after the installment of the heat exchanger in front of the evaporator

2. The second reason is, when using low enthalpy waters or waste waters as the LHS higher temperatures can be reached. For air, the maximum theoretical temperature is equal to the desired temperature of the home's interior.
3. Changing LHS would also result in lower maintenance, service costs and with simplifications in the unit construction, because the heat pump could be connected with a hot water storage tank and installed as a one unit inside the house.

4.2 Design assumptions

The selection of heat exchanger depends on several factors. The most important are costs, dimensions and weight, materials, pumping power, heat transfer rate and pressure drop. In the case of this study some of them, e.g. costs, heat transfer rate and pressure drop are very important and others, such as weight, color and volume are not equally important. Factors and assumptions which were under consideration during the design process are:

1. The smallest pressure drop on the CO₂ side. Complicated geometry on the CO₂ side could cause a large pressure drop, which would lead to a temperature drop. Since it is desirable to obtain the highest temperature at the exit from heat exchanger, special care should be given to simplify the geometry.
2. The pressure at the entrance to the evaporator is about 4 MPa (in calculations 3,9 MPa is assumed). With such high pressure we should be sure that no leaks are possible.

3. This is a gas-to-liquid heat exchanger, so a large heat transfer area should be expected
4. The most important factor since the ancient Phoenicians discovered money is almost always just that, money. It must be taken into consideration that the final product must be competitive on the market, so elements must be relatively easy to manufacture and consist of inexpensive materials.

Taking these points into consideration a shell and tube design was chosen. Point 3 would suggest a plate heat exchanger since it has a larger heat transfer area in the same volume. It will be shown later that dimensions are a limiting factor, but are not decisive.

4.3 General idea – shell and tube heat exchanger

CO₂ is located at the tube side. There are several reasons to do this. Most important are higher pressure losses on the shell side (anyway impossible to be calculated with high accuracy).

The tube side should have the simplest geometry as possible (circular straight tubes) and as large an area as possible (which means many pipes with the smallest available diameter). Also, tube walls should be made of inexpensive, highly thermally conductive material. The shell side, where water flows, should be well isolated to minimize heat losses to the ambient.

As shown in figure 27 the heat exchanger should be divided into two parts: the evaporator and the superheater. This will be an artificial division made to simplify calculation and to adjust locations in heat exchanger to lower indexes that will be used in further calculations. In reality this will be one part.

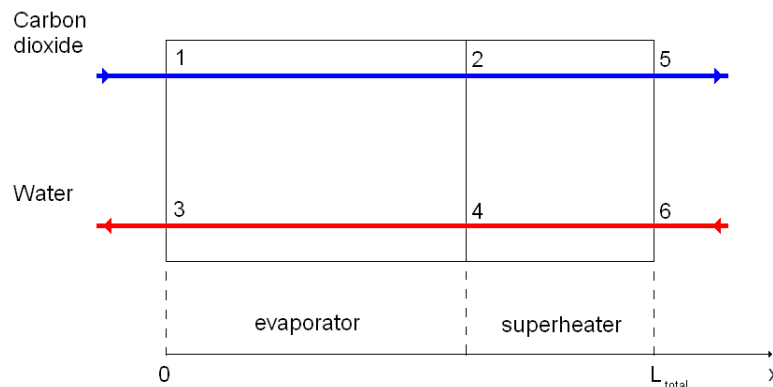


Figure 27 Schema of the heat exchanger showing an imaginary division for two parts: the evaporator and the superheater. Numbers are used as lower indexes and they indicate boundary points in the heat exchanger.

4.4 Design conditions – boundary conditions (temperature, pressure and enthalpy)

The heat exchanger is designed as a counter flow shell and tube. One of goals during the design process was to calculate the optimal length of the heat exchanger. To start the calculations, boundary conditions were needed: values at the beginning and at the end, based on data which was found, assumed or concluded. It will be shown in this work how to calculate the maximum heat exchanger's length for the highest value of temperature of the LHS.

4.4.1 Water side

LHS temperature T_6 is assumed to range from 15°C to 40°C. This will be the temperature of water flowing into the heat exchanger. At the outlet of the heat exchanger the temperature value of $T_3=7^\circ\text{C}$ (temperature almost 3 degrees above CO_2 evaporation temperature) was assumed. Enthalpy at the inlet and outlet was calculated in RefProp (E.W. Lemmon, M.L. Huber, M.O. McLinden, 2007). Water is dependent (since it is in liquid state) with a good approximation on temperature only (table 5) and can be calculated by multiplying temperature by specific heat, which can be assumed to be constant $c=4.18\frac{\text{kJ}}{\text{kg}\times\text{K}}$: $h_6=168\frac{\text{kJ}}{\text{kg}}$ and $h_3=30\frac{\text{kJ}}{\text{kg}}$.

Table 5 Table shows independence of enthalpy on pressure

Temperature (°C)	Pressure (bar)	Volume (m ³ /kg)	Enthalpy (kJ/kg)	Entropy (kJ/kg-K)
40	1	0,001008	167,62	0,572
40	2	0,001008	167,70	0,572
40	3	0,001008	167,79	0,572
40	4	0,001008	167,88	0,572
40	5	0,001008	167,97	0,572

4.4.2 CO₂ side

Since pressure at the inlet to the compressor (4 MPa) is limited, it was assumed that the evaporation process starts at point A (figure 23), where p_1 is 3.9 MPa and quality about $x_1=5\%$. For those values it was possible to find in RefProp (E.W. Lemmon, M.L. Huber, M.O. McLinden, 2007) $t_1=4.3^\circ\text{C}$ and enthalpy $h_1=220\frac{\text{kJ}}{\text{kg}}$. For incoming water (with temperature 40°C) the temperature of carbon dioxide at the outlet of the heat exchanger $t_5=37^\circ\text{C}$ is presumed. Pressure drop (in chapter 4.8) is insignificant in the heat exchanger (on the CO_2 side) and will not exceed 1 bar, so 1 bar drop was assumed and calculated in Refprop (E.W. Lemmon, M.L. Huber, M.O. McLinden, 2007) $h_5=479\frac{\text{kJ}}{\text{kg}}$.

4.5 Mass flows

To calculate the mass flow of CO₂ we need to estimate a few parameters of the air-to-water EcoCute model when its compressor works with maximum lower pressure - 4 MPa. Since the heat pump with changed LHS will work with this pressure all the time, the mass flow which we will obtain will be the same and constant in a new model. According to the Sanyo catalogue 2006 (SANYO, 2006) at ambient temperature equal to 20°C (at this temperature the compressor works with a pressure of 4 MPa) LHS delivers $\dot{Q}_{old} = 3.3$ kW. Enthalpy at point 1 has already been calculated (in chapter 4.4.2) and enthalpy in point 2 is approximately $445 \frac{kJ}{kg}$.

Estimation of this value was based on the following assumptions:

- Pressure was equal to 3.8MPa (around 1 bar drop)
- Temperature was 5-10°C lower from ambient temperature

Having this information:

$$\dot{m}_{CO2} = \frac{\dot{Q}_{old}}{h_2 - h_1} = \frac{3.3 \text{ kW}}{445 \frac{kJ}{kg} - 220 \frac{kJ}{kg}} = 0.0147 \frac{kg}{s}$$

It is assumed that energy contained in water is transferred in the form of heat only to carbon dioxide. There are no heat losses to the surrounding, so minimum water mass flow is:

$$\begin{aligned}\dot{Q}_{water} &= \dot{Q}_{CO2} \\ \dot{Q}_{CO2} &= \dot{m}_{CO2}(h_5 - h_1) = 3.78 \text{ kW} \\ \dot{m}_{water}(h_6 - h_3) &= \dot{m}_{CO2}(h_5 - h_1) \\ \dot{m}_{water} &= \frac{\dot{Q}_{CO2}}{(h_6 - h_3)} = 0.0274 \frac{kg}{s}\end{aligned}$$

4.6 Capillary tubes

Tubes should be made of copper since it has the highest thermal conductivity (from all easily accessible and cheap materials) equal to $401 \frac{W}{m \times ^\circ C}$ (table 6). They also should have the smallest possible OD (outside diameter) to enlarge the total heat transfer area (because of higher amount of tubes that could be installed in the same volume) and the largest possible ID (internal diameter) to decrease fluid velocity (to receive smaller pressure losses) and reduce R_w (thermal resistance of the wall). From all producers “Quen Cheer Industrial Co., Ltd.” was chosen, which provides suitable tubes with ID=0.7mm and OD=1.2mm (Quen Cheer Industrial Co., 2009). A cross section area of this tube is:

$$A_{tube} = \frac{\pi \times ID^2}{4} = 0.00000038465 \text{ m}^2$$

Table 6 Thermal conductivities of chosen materials

Thermal conductivity at 293K	
Material	Value

Diamond	2300
Silver	429
Copper	401
Gold	317
Aluminium	237
Iron	80,2
Mercury (liquid)	8,54
Glass	0,78
Brick	0,72
Water	0,613
Human skin	0,37
Oak	0,17
Helium	0,152
Soft rubber	0,13
Glass fiber	0,043
Air	0,026

Tubes were made of electrolytic tough pitch copper, UNS C11000, OSO25 (Quen Cheer Industrial Co., 2009). This alloy has a copper fraction higher than 99.9% and yield tensile strength of 76 MPa (MatWeb, 2009). For this material, ID, OD, the maximum stress concentration factor (equal to 3), pressure inside (3.9 MPa) and outside the tube (0.2 MPa) - maximum allowed pressure inside the pipe is equal to 14.36 MPa (Brzozka, 1979). This value is 3.68 times higher than the pressure inside the tube, which is an important safety factor, since tensile strength will be lower at the bends and at the entrance and exit of the tube. In fact, wall thickness could be probably 1 mm lower, but this would lead to serious risk and would almost not affect heat transfer.

4.7 Flow division

Pipes with CO₂ at the inlet and outlet of the evaporator have an internal diameter $D_{pipe} = 6.4$ mm (SANYO, 2005), so a cross section area of this tube is:

$$A_{pipe} = \frac{\pi \times D_{pipe}^2}{4} = 0.0000321536 \text{ m}^2$$

CO₂ specific volume at the point 1 is $0.001443 \frac{\text{m}^3}{\text{kg}}$ (table 3). Having these values, we can calculate the speed of the working fluid before the inlet to the evaporator:

$$w_{pipe} = \frac{V_{s,pipe} \times m_{CO2}}{A_{pipe}} = \frac{0.0147 \frac{\text{kg}}{\text{s}} \times 0.001443 \frac{\text{m}^3}{\text{kg}}}{0.005024 \text{ m}^2} = 0.655 \frac{\text{m}}{\text{s}}$$

Flow at the entrance to the evaporator was divided with use of 6 rows of Y-junctions. This approach will guarantee that 64 equal mass flows will be obtained.

$$\dot{m}_{tube} = \frac{\dot{m}_{CO2}}{64} = 0.000458 \frac{\text{kg}}{\text{s}}$$

From the continuity equation we can calculate the tube's working fluid velocity:

$$\begin{aligned}
64 \times \dot{m}_{tube} &= \dot{m}_{CO2} \\
64 \times w_{tube} \times A_{tube} \times \rho_{CO2} &= w_{pipe} \times A_{pipe} \times \rho_{CO2} \\
w_{tube} &= \frac{w_{pipe} \times A_{pipe}}{64 \times A_{tube}} = 0.855 \frac{m}{s}
\end{aligned}$$

4.8 Pressure drop

There are two types of pressure losses: major and minor. The first type is connected with fluid friction during flow in a pipe and the second one with changes in the geometry of the flow.

4.8.1 Major losses

In the heat exchanger, 1 bar pressure loss was assumed (assumption turned out to be quite accurate, so there was no need to make iterations). In the RefProp (E.W. Lemmon, M.L. Huber, M.O. McLinden, 2007) isoproperty table (Table A) for carbon dioxide was prepared, where a constant pressure of 3.8 MPa was held and temperature varied from 4°C to 37°C, with step 0.5°C. For every single row velocity, Reynolds number (Re) and friction factor f were calculated from formulas below:

$$\begin{aligned}
w_i &= \frac{V_{s,i} \times m_{CO2}}{A_{pipe}} \\
Re &= \frac{ID \times w_i}{v_i} \\
f &= (0.79 \times \ln Re - 1.64)^{-2}
\end{aligned}$$

Where V_s is specific volume, w velocity and v kinematic viscosity. Lower index i stands for every next row in table A (appendix A).

Reynolds number is a dimensionless quantity; it is a ratio of inertia forces to viscous forces and it differentiates laminar flow from a turbulent one. Pure laminar flow ends at $Re = 2300$ and pure turbulent flow starts above 10000 (Cengel, 2003). Between the two extremes the so-called transitional flow exists.

Friction factor is also a dimensionless value. In the case of this study it is calculated from the explicit first Petukhov equation (Petukhov, 1970). It is valid for Re between 3000 and 5,000,000; and Prandtl number (Pr – described in chapter 4.10.1) between 0.5 and 2000. Both conditions are fulfilled.

After calculation of values for particular rows the average value of Reynolds number (26608), velocity ($6.57 \frac{m}{s}$), friction factor (0.024775) and density ($91.3 \frac{kg}{m^3}$) were calculated. Having these parameters, the value of major losses was obtained for length $L = 1m$:

$$\Delta p_{major} = \frac{L \times \rho_{CO2 \text{ average}} f_{CO2 \text{ average}} w_{CO2 \text{ average}}^2}{ID \times 2} = 0.7 \text{ bar}$$

4.8.2 Minor losses

In the designed heat exchanger there are five places on the working fluid side where losses will occur due to changes in geometry. They were included in calculations, but it will be proved that they are insignificant and can be omitted.

Y-junction losses

Working fluid, on its route, will go through six rows of divergent and six rows of convergent Y-junctions (figure 28). Junctions have the same internal diameters at the inlet and outlet. Pressure losses in divergent junctions can be calculated from the equation:

$$\Delta p_{Y-jun,d} = \frac{f_{Y-jun,d} \rho_1 w_{tube}^2}{2} = \frac{0.067 \times 693 \frac{kg}{m^3} \times 0.855^2 \frac{m}{s}}{2} = 0.00017 \text{ bar}$$

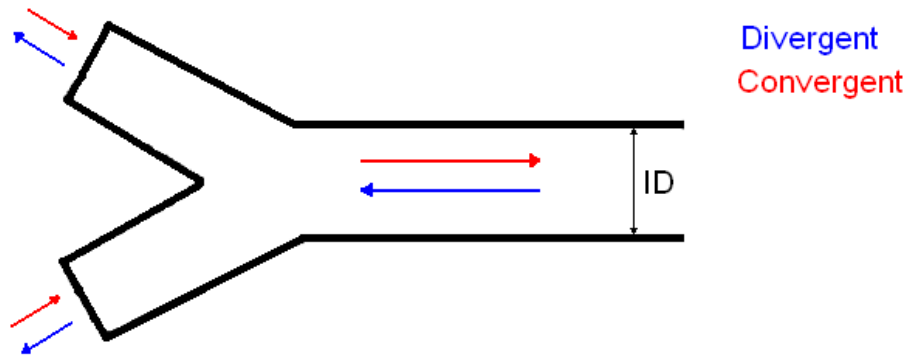


Figure 28 Y-junction

Value of ρ_1 was found in RefProp (E.W. Lemmon, M.L. Huber, M.O. McLinden, 2007). Value of $f_{Y-jun,d}$ was found in VDI Heat atlas (VDI, 1991). It is taken for a larger diameter because there is no data for such small diameters, but the highest value was taken and the result is very small. If the f was 10 larger it would not influence final design, since pressure drop would still be negligible.

Table 7 Friction factors for symmetrical y-junctions (VDI, 1991)

Values of friction factor for symmetrical y-junctions		
Internal diameter	Divergence	Convergence
350 mm	0.067	0.17
140 mm	0.035	0.08

Value for the convergent case can be calculated in the same way:

$$\begin{aligned}\Delta p_{Y-jun,c} &= \frac{f_{Y-jun,c} \rho_5 w_5^2}{2} = \\ &= \frac{0.035 \times 693 \frac{kg}{m^3} \times 0.855^2 \frac{m}{s}}{2} = 0.00377 \text{ bar}\end{aligned}$$

Losses connected with a change of a diameter

After the divergent y-junctions there is a change in diameter because it is necessary to switch from the diameter of the pipe to the diameter of the capillary tube. To receive the smallest pressure losses the cross section of the pipe should continuously be reduced (figure 29), with $\alpha < 40^\circ$ (above this value the stream of the fluid in the jet can separate (VDI, 1991)). If this condition is fulfilled then this equation can be used:

$$\begin{aligned}\Delta p_{reduction} &= \frac{0.04 \times \rho_1 \times w_{tube}^2}{2} = \\ &= \frac{0.035 \times 693 \frac{kg}{m^3} \times 0.855^2 \frac{m}{s}}{2} = 0.000088 \text{ bar}\end{aligned}$$

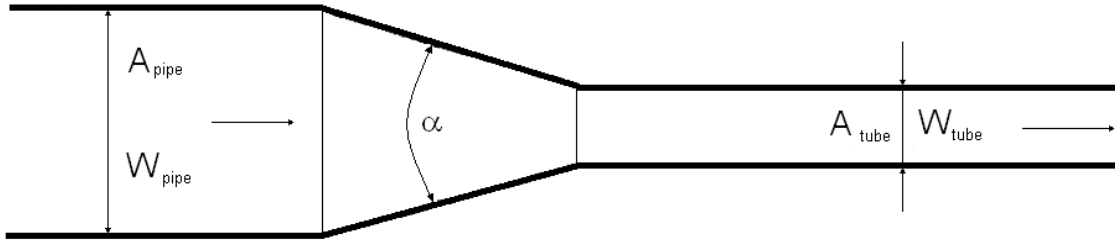


Figure 29 A continuous cross section area reduction

At the end of the superheater we have the opposite situation – continuous widening of the cross section (figure 30). To minimize pressure losses β should be equal to 5° (VDI, 1991). For $\beta = 5^\circ$ and $\frac{ID}{D_{pipe}}$ we can estimate the value of the friction factor ($f_{widening}$), which is equal to 0.15 (VDI, 1991).

$$\begin{aligned}\Delta p_{widening} &= \left(1 - \frac{A_{tube}}{A_{pipe}}\right) \frac{f_{widening} \rho_5 w_5^2}{2} = \\ &= \left(1 - \frac{0.00000038465 \text{ m}^2}{0.0000321536 \text{ m}^2}\right) \frac{0.15 \times 80 \frac{kg}{m^3} \times 7.45^2 \frac{m^2}{s^2}}{2} = 0.0216 \text{ bar}\end{aligned}$$

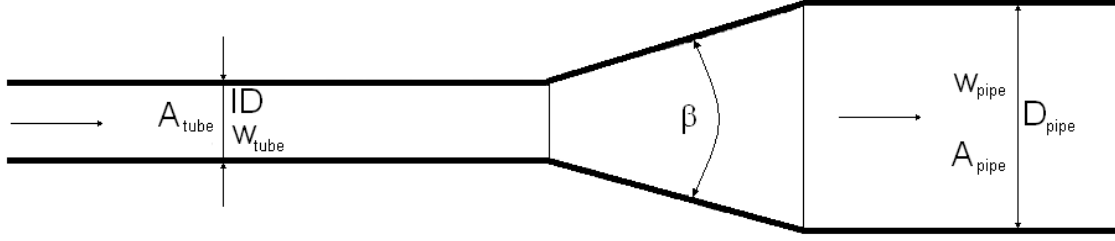


Figure 30 A continuous cross section area widening

Bend losses

It was calculated (in chapter 4.10 below) that the heat exchanger will be too long to fit in the heat pump casing (figure 23), so it must be bent in half. Friction factor was taken based on fig.15 on page Lc 5 in the VDI Heat Atlas (VDI, 1991). There is no value which might match, so the highest probable value was chosen - 0.4. Since the bend is in the middle of heat exchanger, average values of density and velocity will be used in the calculations (refer to chapter 4.8.1).

$$\begin{aligned}\Delta p_{bend} &= \frac{f_{bend} \times \rho_{CO2 \text{ average}} \times w_{CO2 \text{ average}}^2}{2} = \\ &= \frac{0.4 \times 91.3 \frac{kg}{m^3} \times 6.57^2 \frac{m}{s}}{2} = 0.007886 bar\end{aligned}$$

4.8.3 Pressure losses - conclusions

Total pressure drop for one meter of the evaporator is:

$$\begin{aligned}\Delta p_{CO2} &= \Delta p_{major} + \Delta p_{minor} = \\ &= \Delta p_{major} + \Delta p_{bend} + 6 \times \Delta p_{widening} + 6 \times \Delta p_{reduction} + \Delta p_{Y-jun,c} + \\ &\quad + \Delta p_{Y-jun,d} = 0.736622 bar\end{aligned}$$

Some additional pressure losses may occur due to connections between parts, but the total pressure drop should not exceed 1 bar per 1 meter of the evaporator length (which is about the length of our heat exchanger - calculated in chapter 4.9). It means that at the end of the evaporator, pressure will be equal to 3.8 MPa and the temperature at 37°C. Enthalpy for these values is $479 \frac{kJ}{kg}$.

4.9 Length of the heat exchanger

Length of the heat exchanger L_{total} is a crucial parameter. Too short a design will not allow reaching the desired temperature at the end and too long will cause additional pressure losses, which will lead to a decrease of enthalpy. The aim is to be as close to the actual value as possible. To achieve this target the amount of heat exchanged in both parts of the heat exchanger must be calculated.

$\dot{Q} = \dot{m} \times \Delta h$, so the rate of heat transfer in the evaporator is:

$$\dot{Q}_{evaporator} = \dot{m}_{CO2} \times (h_2 - h_1) = 0.0147 \frac{kg}{s} \times \left(428 \frac{kJ}{kg} - 220 \frac{kJ}{kg} \right) = 3.05 kW$$

h_2 is the value of enthalpy (table 8), where the process of evaporation ends ($x=1$). It means that to obtain this value it is sufficient to know the pressure at the end of the evaporator (0.5 bar pressure drop was presumed).

Table 8 Parameters for saturation pressure equal to 3.85 MPa

Temperature (°C)	Pressure (MPa)	Liquid Density (kg/m ³)	Vapor Density (kg/m ³)	Liquid Enthalpy (kJ/kg)	Vapor Enthalpy (kJ/kg)
3,81	3,85	903,78	110,30	209,47	428,39

Similarly, the rate of heat transfer in the evaporator can be found:

$$\dot{Q}_{superheater} = \dot{m}_{CO_2} \times (h_5 - h_2) = 0.0147 \frac{kg}{s} \times \left(479 \frac{kJ}{kg} - 428 \frac{kJ}{kg} \right) = 0.748 kW$$

4.9.1 Length of the evaporator

In every heat exchanger the rate of heat transfer depends on:

- Total heat transfer resistance R_t
- Logarithmic mean temperature difference Δt_{lm}

$$\dot{Q}_{evaporator} = \frac{\Delta t_{lm}}{R_{t, evaporator}}$$

Total heat transfer resistance

Total resistance is the sum of all resistances between two boundary temperatures (in our case between CO₂ temperature and water temperature, see figure 31). To get $R_{t, evaporator}$ we will have to calculate the sum of average values of three resistances:

- Tube wall resistance R_{wall}

This resistance is easy to obtain, because it depends only on the thermal conductivity of copper, wall thickness and area:

$$R_{wall} = \frac{\ln \left(\frac{OD}{ID} \right)}{2 \times \pi \times L \times k_{copper}}$$

$$R_{wall} \times L = \frac{\ln \left(\frac{OD}{ID} \right)}{2 \times \pi \times k_{copper}} = 0.000214 \frac{^{\circ}C \times m}{W}$$

- Convective resistance on CO₂ side $R_{CO_2, conv, evap}$

Is described by the equation:

$$R_{CO_2, conv, evap} = \frac{1}{A_{evaporator} \times \alpha_{CO_2, ave, evap}}$$

$$R_{CO2,conv,evap} \times L = \frac{1}{ID \times \pi \times \alpha_{CO2,ave,evap}}$$

The inside area of a cylinder is known to be $A_{evaporator} = \pi \times ID \times L$, but it is very difficult to obtain $\alpha_{CO2,ave,evap}$. It will be estimated later.

- Convective resistance on water side $R_{H2O,conv,evap}$

This value is obtained in the same way, as $R_{CO2,conv,evap}$ and also a separate chapter will be devoted to the calculations of the convective coefficient.

$$R_{H2O,conv,evap} = \frac{1}{A_{evaporator} \times \alpha_{H2O,ave,evap}}$$

$$R_{H2O,conv,evap} \times L = \frac{1}{OD \times \pi \times \alpha_{H2O,ave,evap}}$$

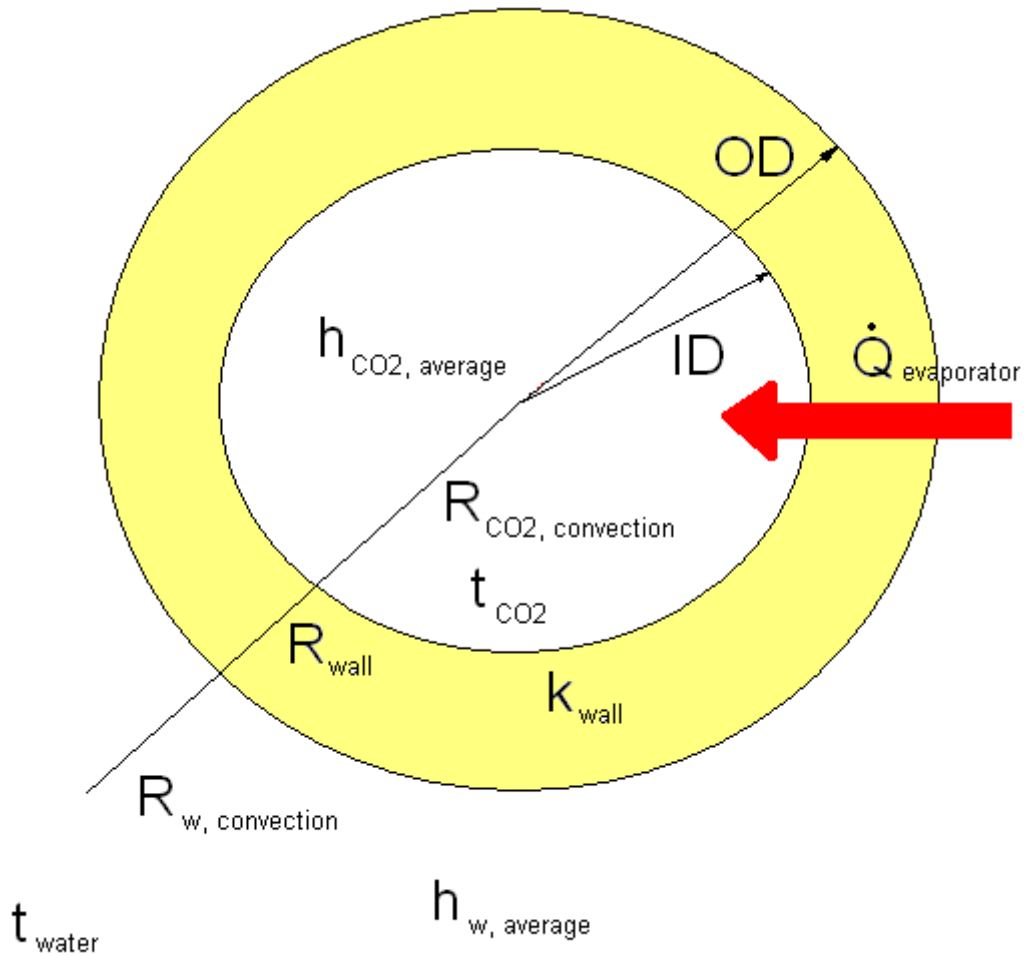


Figure 31 Resistance concept for a cylindrical tube

Logarithmic mean temperature difference

The exact representation of the average temperature difference between the two fluids in the heat exchanger is expressed as:

$$\Delta t_{lm} = \frac{\Delta t_a - \Delta t_b}{\ln \left(\frac{\Delta t_a}{\Delta t_b} \right)}$$

Where Δt_a and Δt_b represent temperature difference at both sides of a heat exchanger (figure 32).

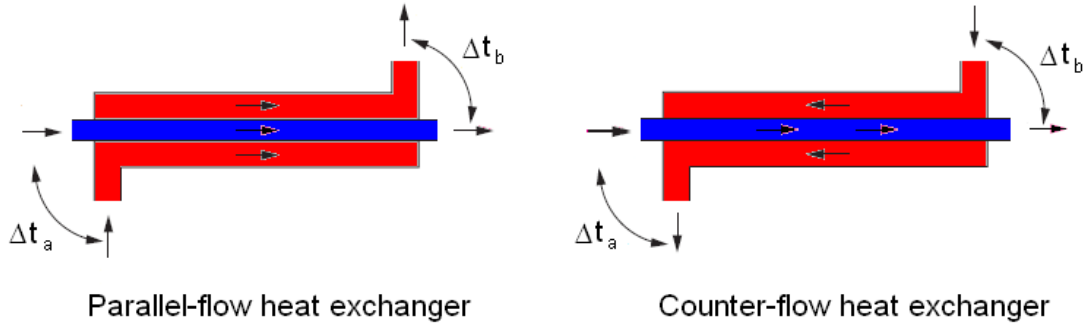


Figure 32 Parallel and counter flow heat exchanger

We know that in the evaporator the CO_2 temperature is constant ($t_1 = t_2 = 4.3^\circ\text{C}$). We know also the temperature of water coming out of the heat exchanger ($t_3 = 7^\circ\text{C}$). Since we assume that all heat is transferred from water to CO_2 , we can easily calculate t_4 :

$$t_4 = \frac{\dot{Q}_{\text{evaporator}}}{c \times \dot{m}_{\text{water}}} + t_3 = 33.6^\circ\text{C}$$

Having all these values, we can calculate the logarithmic mean temperature difference for our case:

$$\Delta t_{lm} = \frac{(4.3^\circ\text{C} - 7^\circ\text{C}) - (4.3^\circ\text{C} - 33.6^\circ\text{C})}{\ln \left(\frac{4.3^\circ\text{C} - 7^\circ\text{C}}{4.3^\circ\text{C} - 33.6^\circ\text{C}} \right)} = 11.16^\circ\text{C}$$

Estimation of the convection coefficient on the working fluid side in the evaporator

$\alpha_{\text{CO}_2, \text{ave, evap}}$ can be estimated rather than calculated because of the complicated two phase flow, the nature of which is not well known. Therefore, all calculations in the two-phase region are based on empirical equations (if they exist). At the entrance to the evaporator, the working fluid is in the liquid phase and at the end of the evaporator only in gas phase. The difficulty in estimating the convection coefficient occurs in between, where we have many flow patterns (figure 33). They vary with the quality of the working fluid in the evaporator and are caused by the phase change and effects of gravity. They strongly affect the convective heat transfer coefficient. According to the VDI heat atlas (VDI, 1991) convective heat transfer coefficient $\alpha_{\text{CO}_2, \text{ave, evap}}$ in horizontal tube evaporator is described by following equation:

$$\alpha_{CO2,ave,evap} = \sqrt[3]{\alpha_{CO2,evap,conv}^3 + \alpha_{CO2,evap,nuc}^3}$$

Where $\alpha_{CO2,evap,conv}$ describes the average convective boiling and $\alpha_{CO2,evap,nuc}$ the average nucleate boiling convection coefficient.

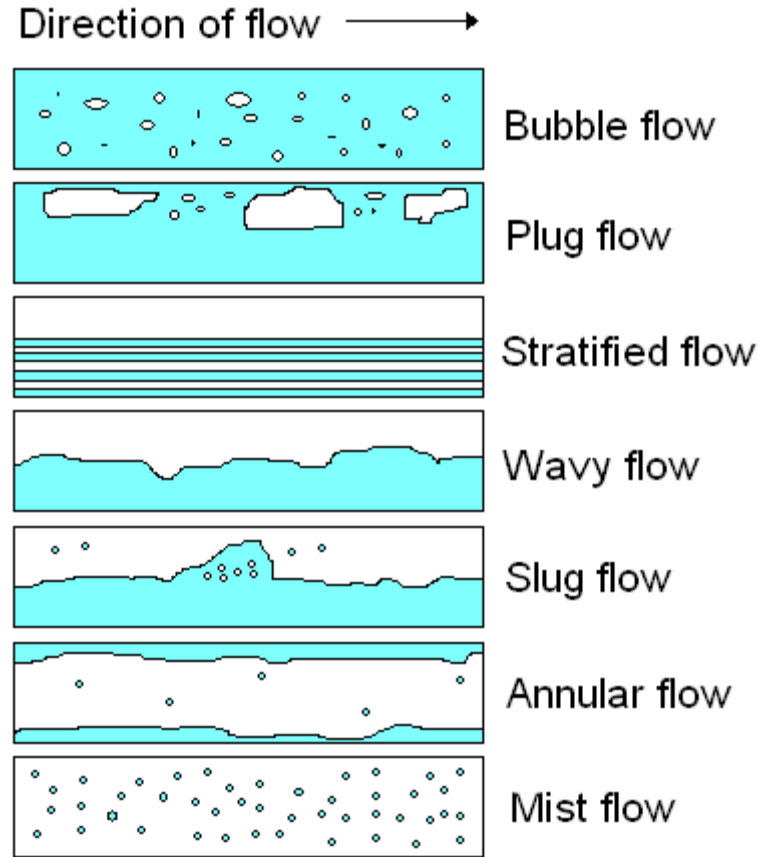


Figure 33 Two phase patterns in horizontal tubes

Convection flow boiling in horizontal tubes

$\alpha(x_i)_{CO2,evap,conv}$ is the convective heat transfer coefficient in convective boiling regime, in the horizontal tube, at the point with quality x_i . It is described by the equation below:

$$\alpha(x_i)_{CO2,evap,conv} = \left\{ \left[\left((1 - x_i)^{1.5} + 1.2 \times x_i^{0.4} \times (1 - x_i)^{0.01} \times \left(\frac{p_z}{p_2} \right)^{0.37} \right)^{-2.2} + \left(\frac{\alpha_2}{\alpha_1} \times x_i^{0.01} \times \left(1 + 8 \times (1 - x_i)^{0.7} \times \left(\frac{p_z}{p_2} \right)^{0.67} \right) \right)^{-2} \right]^{-0.5} \right\} / \alpha_1$$

Where ρ_1 and ρ_2 are carbon dioxide mass densities at points 1 and 2, respectively. α_2 and α_1 are heat transfer convection coefficients at the same points. Both of them were calculated in the following way:

$$\alpha_{1 \text{ or } 2} = \frac{k_{1 \text{ or } 2} \times Nu_{1 \text{ or } 2}}{ID}$$

Where k – the thermal conductivity, is a property of the working fluid and was found in RefProp (E.W. Lemmon, M.L. Huber, M.O. McLinden, 2007). Nu – Nusselt number is a dimensionless number, made to combine the variables, to reduce their total amount. It can be obtained with high accuracy from the following equation (Petukhov, 1970):

$$Nu_{1 \text{ or } 2} = \frac{\left(\frac{f_{1 \text{ or } 2}}{8}\right) \times (Re_{1 \text{ or } 2} - 1000) \times Pr_{1 \text{ or } 2}}{1 + 12.7 \times \left(\frac{f_{1 \text{ or } 2}}{8}\right)^{0.5} \times (Pr_{1 \text{ or } 2}^{\frac{2}{3}} - 1)}$$

This equation is valid for Re in range of 3000 to 5000000 and Pr from 0.5 to 2000. Both conditions are fulfilled.

The Prandtl number is a dimensionless number defined as the molecular diffusivity of momentum to the molecular diffusivity of heat. It is the fluid property at a specified point and was calculated in Refprop (E.W. Lemmon, M.L. Huber, M.O. McLinden, 2007). To obtain Re and f the equations below were used:

$$w_{1 \text{ or } 2} = \frac{V_{s,1 \text{ or } 2} \times \dot{m}_{CO2}}{A_{tube}}$$

$$Re_{1 \text{ or } 2} = \frac{ID \times w_{1 \text{ or } 2}}{\nu_{1 \text{ or } 2}}$$

$$f_{1 \text{ or } 2} = (0.79 \times \ln Re_{1 \text{ or } 2} - 1.64)^{-2}$$

All data needed for these equations and results for velocity, Reynolds number, Nusselt number, friction factor and finally convective heat transfer coefficients for point 1 and 2 are listed in table 9.

Table 9 Properties of points with quality 0 and 1 for pressure 3.85 MPa

	Therm. Cond. (mW/m-K)	Kin. Visc (cm ² /s)	Prandtl	Velocity (m/s)	Re	f	Nu	alpha (W/m ² -K)
Liquid	105,77	0,00103	2,3493	0,8554	5830	0,036	31,028	4688,26
Vapor	21,092	0,00138	1,4919	5,5451	28140	0,024	101,8	3067,47

Having α_2 and α_1 and densities at point 1 and 2 it was possible to solve the equation for $\alpha(x_i)_{CO2, \text{evap}, \text{conv}}$. The equation was solved for quality in a range from 0.05 to 1. It means that $i=21$ results were obtained (table 10 and figure 34), so to get $\alpha_{CO2, \text{evap}, \text{conv}}$ the

average value of $\alpha(x_i)_{CO2, evap, conv}$ was calculated. In our the case average convection coefficient in the convective flow boiling regime is equal to $\alpha_{CO2, evap, conv} = 10806 \frac{W}{m^2 \times ^\circ C}$.

Table 10 Convection coefficient values in the two phase region for different qualities

Temperature (°C)	Pressure (MPa)	Quality (kg/kg)	alpha (W/m2-K)
3,8106	3,85	0	4688,262
3,8106	3,85	0,05	8499,524
3,8106	3,85	0,1	9511,096
3,8106	3,85	0,15	10157,48
3,8106	3,85	0,2	10623,79
3,8106	3,85	0,25	10979,23
3,8106	3,85	0,3	11258,41
3,8106	3,85	0,35	11481,4
3,8106	3,85	0,4	11660,83
3,8106	3,85	0,45	11805,03
3,8106	3,85	0,5	11919,46
3,8106	3,85	0,55	12007,37
3,8106	3,85	0,6	12069,91
3,8106	3,85	0,65	12105,79
3,8106	3,85	0,7	12110,05
3,8106	3,85	0,75	12071,23
3,8106	3,85	0,8	11964,37
3,8106	3,85	0,85	11731,77
3,8106	3,85	0,9	11218,6
3,8106	3,85	0,95	9875,782
3,8106	3,85	1	3067,471
average			10805,93

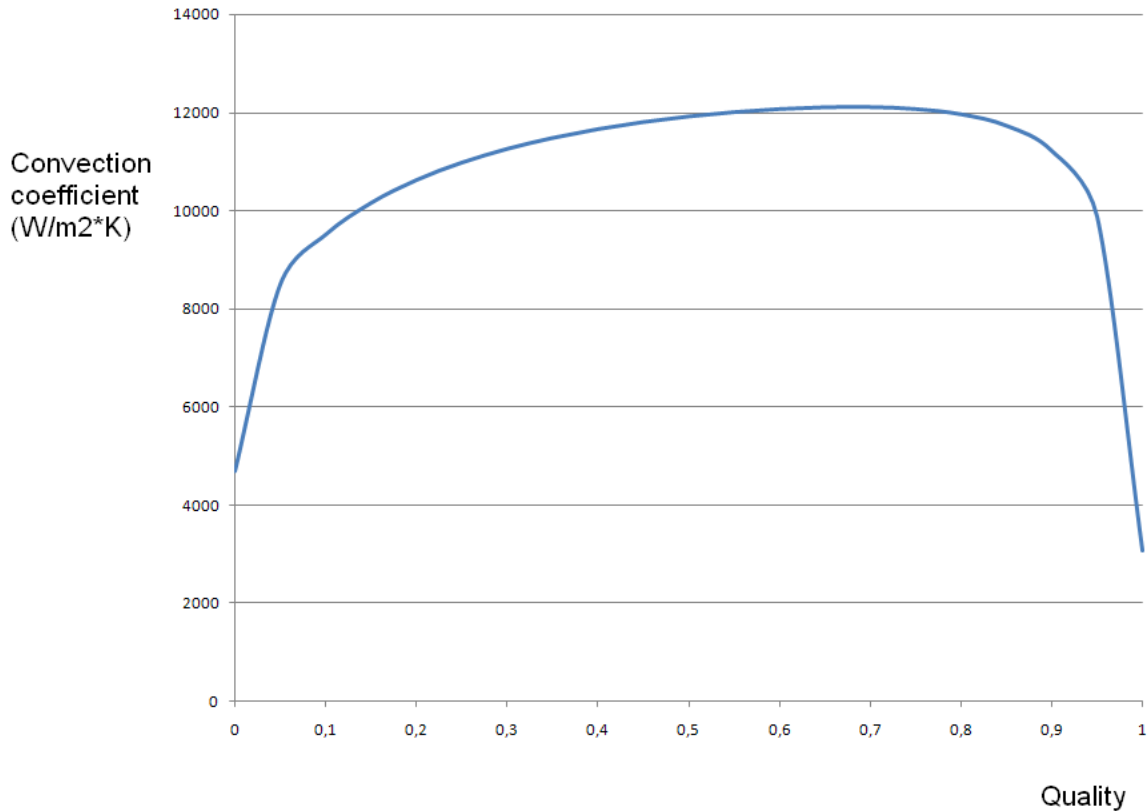


Figure 34 Convection coefficient values in the two phase region for different qualities

Nucleate flow boiling in horizontal tubes

According to literature, no formula exists that can be used to calculate $\alpha_{CO2,evap,nucl}$ unless for bigger tube diameters, but this problem can be overcome.

As mentioned before, the thermal resistance to convection is inversely proportional to the convection coefficient, so if the convection coefficient is high, then thermal resistance is low and heat can pass through more quickly. It simply means that if the convection coefficient is high then the heat exchanger can be shorter. It is necessary to remember that the convection heat transfer coefficient $\alpha_{CO2,ave,evap}$ in the horizontal tube evaporator is described by following equation:

$$\alpha_{CO2,ave,evap} = \sqrt[3]{\alpha_{CO2,evap,conv}^3 + \alpha_{CO2,evap,nucl}^3}$$

Under the root is the sum of two elements. If we presume one of them - $\alpha_{CO2,evap,nucl}$ to be equal to 0 then we will need a longer heat exchanger. This distance will be a few cm and can be considered a safety factor which will not affecting the pressure drop.

Estimation of the convection coefficient on water side in the evaporator

To obtain the value of $\alpha_{H2O,ave,evap}$ an isoproperty table was created (table B – appendix A), where a constant pressure of 2 bar was held and temperature was changed from 7°C to 33.5°C with step 0.5°C. Water velocity in the duct was calculated before being disturbed (by tube banks with CO₂) with the equation:

$$w_j = \frac{V_{s,j} \times \dot{m}_{water}}{A_{waterduct}}$$

Lower index j is the number of a row in table B (starting with 1 and ending at 54). According to the continuity equation (chapter 4.8), when fluid changes its flow cross section area, its velocity also changes. When water enters a region with tube banks the cross section of the flow area decreases and velocity has to be recalculated from the formula for the tube banks' in-line arrangement (figure 35):

$$w_{j,max} = \frac{w_j \times S_t}{S_t - S_l}$$

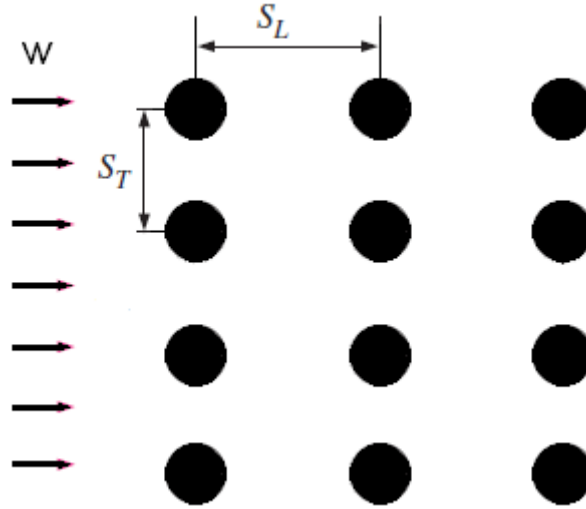


Figure 35 Tube banks in-line arrangement

Where S_t and S_l are distances between tubes (see figure 35). The next step was the calculation of Re :

$$Re_j = \frac{OD \times w_{j,max}}{v_j}$$

Then the following equation for Nusselt number was used (Zukauskas, 1987):

$$Nu_j = 0.9 \times Re_j^{0.4} \times Pr_j^{0.36} \times \left(\frac{Pr_j}{Pr_{s,j}}\right)^{0.25}$$

This equation is valid for flow over the tube banks with the number of tubes $N > 16$, $Re < 100$, $0.7 < Pr < 500$, so all conditions are fulfilled. Pr_s is Prandtl number for water at tube wall surface temperature.

Convection coefficient for row j was calculated from the equation:

$$\alpha_j = \frac{k_j \times Nu_j}{OD}$$

The average of the convective heat transfer coefficient for water in the evaporator is given as:

$$\alpha_{H_2O,ave,evap} = \frac{\sum_{j=1}^{54} \alpha_j}{54} = 3006 \frac{W}{m^2 \times ^\circ C}$$

Results for the evaporator

Since we have thermal conductivity and two convection coefficients we are able to calculate the length of the evaporator:

$$L = \frac{\dot{Q}_{evaporator} \times \left(\frac{\ln \left(\frac{OD}{ID} \right)}{2 \times \pi \times k_{copper}} + \frac{1}{ID \times \pi \times \alpha_{CO_2,ave,evap}} + \frac{1}{OD \times \pi \times \alpha_{H_2O,ave,evap}} \right)}{\Delta t_{lm}}$$

$$L = 0.615m = 61.5cm$$

4.9.2 Length of the superheater

A similar approach has to be applied to calculate the length of the superheater (l). It means that the average convection coefficients have to be calculated on both the water and CO_2 sides. Thermal conductivity of the tube wall is the same since no changes were made to the geometry of the wall.

Estimation of the convection coefficient on the water side in the superheater

$\alpha_{H_2O,ave,sh}$ has been calculated in the same way as in the case of the evaporator (see chapter 4.8.2) and is equal to $2826 \frac{W}{m^2 \times ^\circ C}$. The only difference was that the temperature range is from $33.5^\circ C$ to $40^\circ C$. All data and calculated values are presented in table B.

Estimation of the convection coefficient on the working fluid side in the superheater

In the superheater on the tube side there is gas only. In Refprop an isoproperty table was made (table A – appendix A), where a constant pressure of 3.8 MPa was held and temperature varied in the range from $4^\circ C$ to $37^\circ C$, with step $0.5^\circ C$. Then velocity, friction factor, and Reynolds number (from equations used in the chapter 4.8.1 with lower index “1 or 2”) were obtained for every row in the table. Having these values, the Nusselt number was calculated for every row from the formula (Gnielinski, 1976) expressed as:

$$Nu_k = \frac{\left(\frac{f_k}{8} \right) \times (Re_k - 1000) \times Pr_k}{1 + 12.7 \times \left(\frac{f_k}{8} \right)^{0.5} \times (Pr_k^{\frac{2}{3}} - 1)}$$

Where k is a value from 1 to 24 and it is a number of every next row. Having the Nusselt number, it was possible to calculate the convection coefficient for every temperature in the table. Finally, the average was calculated:

$$\alpha_{CO_2,ave,sh} = \frac{\sum_{k=1}^{24} \alpha_k}{24} = 2448 \frac{W}{m^2 \times ^\circ C}$$

Result for the superheater

$$l = \frac{\dot{Q}_{superheater} \times \left(\frac{\ln \left(\frac{OD}{ID} \right)}{2 \times \pi \times k_{copper}} + \frac{1}{ID \times \pi \times \alpha_{CO2,ave,sh}} + \frac{1}{OD \times \pi \times \alpha_{H2O,ave,sh}} \right)}{\Delta t_{lm}}$$

$$l = 0.3m = 30cm$$

4.9.3 Final design

The total length of the heat exchanger is the length of the evaporator added to the length of the superheater $L_{total} = l + L = 91.5$ cm. To make sure that no unexpected factors (like scaling) will occur it is recommended to increase the heat exchanger length to 1 m. The new casing (chapter 3.2.3) will be 0.6 m long, so to fit the heat exchanger there, it will be necessary to bend it in half. To allow changes to the original design, a program based on the assumptions presented in the previous chapters was created (appendix C).

4.10 Water mass flow at different LHS temperatures

After the length was obtained, all was recalculated once again for the incoming water temperatures: 15°C, 20°C, 25°C, 30°C, 35°C. Of course the new inlet temperature will cause changes in the length. This problem was solved assuming that the length will be held constant (0.9 m) and the inlet water mass flow will vary. Figure 36 shows the relationship between the water mass flow and the water temperature at the inlet (for fixed length of the heat exchanger).

To obtain the maximum performance of the heat pump it would be recommended to install a water pump which will provide the mass flow of water according to figure 36 (or table 11) or slightly higher.

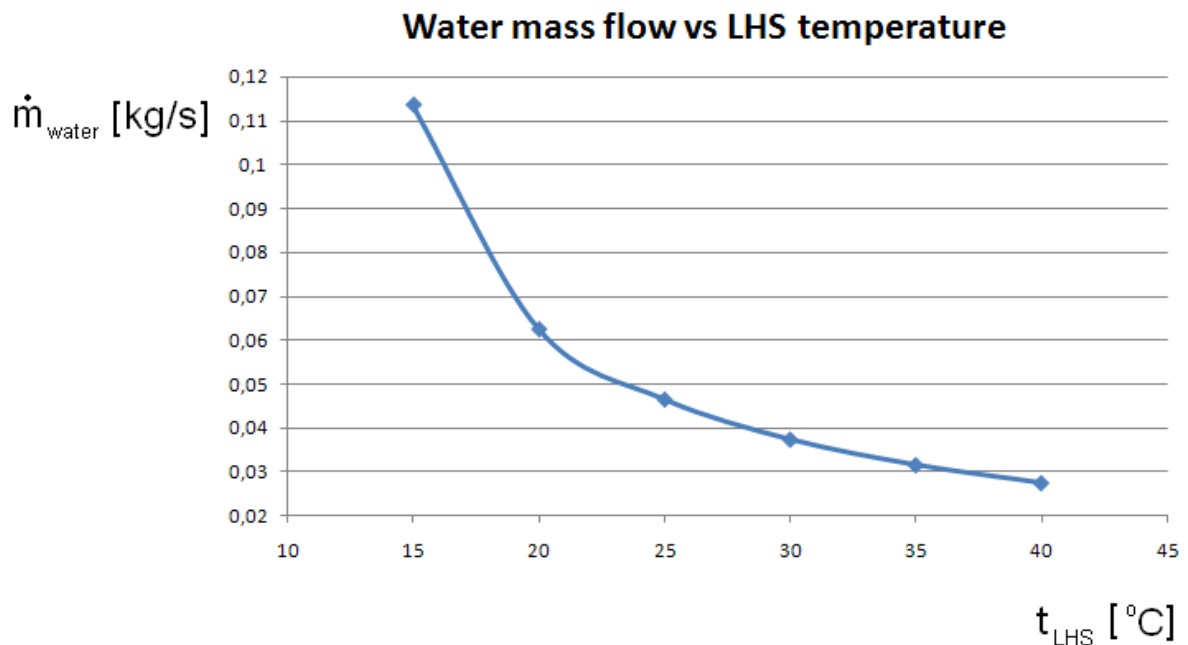


Figure 36 Chart showing how water mass flow depend on ground water temperature

Table 11 Dependence of COP and water mass flow on inlet water temperature

Water temperature [°C]	COP	mass flow [kg/s]
15	3,79	0,1135
20	4,10	0,0625
25	4,54	0,0465
30	5,04	0,0374
35	5,65	0,0316
40	6,42	0,0275

4.11 Influence of the new design on COP

Figure 37 shows how COP changes with inlet water temperature.

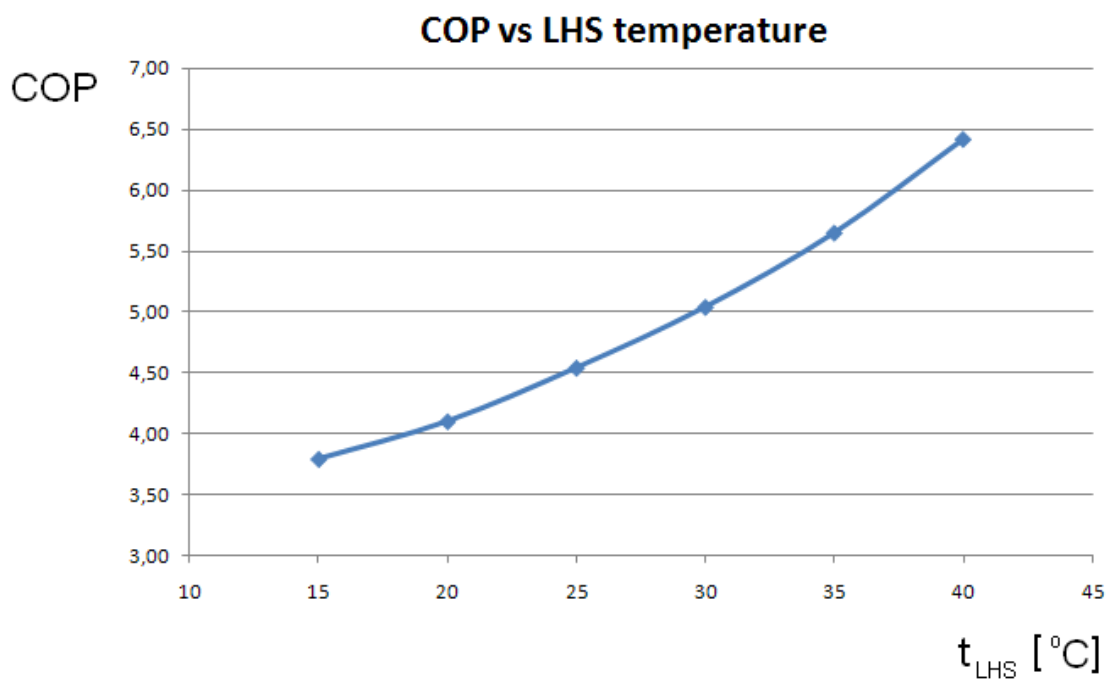


Figure 37 Chart showing how COP depend on ground water temperature

5 CONCLUSIONS

NOTE 1:

The EcoCute CO₂ heat pumps with the new evaporator design could be a perfect alternative for the resistant electric heaters in Iceland, which are used in 9% of Icelandic households. A COP ranging from 3.7 up to 6.5 could provide at least 73% savings of electricity. This means that the government could stop or lower subsidies without affecting end-users.

NOTE 2:

The new design could be sold outside Iceland to places where low enthalpy waters are available in Europe (e.g. Hungary, Turkey) or all around the world, where it could be used with waste water.

NOTE 3:

As traditional refrigerants (CFCs, HCFCs, HFCs) are slowly removed from the market, carbon dioxide seems to be a promising working fluid for the future, because it has ODP=0 and GWP=1, is nontoxic and inflammable and has relatively good physical properties.

NOTE 4:

The new heat pump with the designed evaporator will have high seasonal heating performance, since it utilizes LHS with relatively high, constant temperature.

NOTE 5:

Proper water mass flow should be provided to guarantee maximum performance.

REFERENCES

- ASHRAE. (1993). *Handbook of fundamentals*. Atlanta: ASHRAE.
- Ásmundsson, R. (2008). *Koltvísýringsvarmadæla á Grýtubakka I. ISOR-08032*. Akureyri: ISOR
- BP . (2008). *BP Statistical Review*. London: BP p.l.c.
- BP. (2004). *statistical review of world energy 2004*. London: BP p.l.c.
- Brzozka, Z. (1979). *Wytrzymałosc Materialow*. Warszawa: PWN
- Cengel, Y.A. (2003). *Heat Transfer. A Practical Approach*. New York: McGraw Hill.
- Centre, H. P. (2006). *HeatPumpCentre*. Heat pumps can cut global CO2 emissions: www.heatpumpcentre.org/publ/HPCorder/ViewDocument.aspx?RapportID=451
- CERA. (14. November 2006). *Peak Oil Theory – “World Running Out of Oil Soon” – Is Faulty; Could Distort Policy & Energy Debate*.
<http://www.cera.com/aspx/cda/public1/news/pressReleases/pressReleaseDetails.aspx?CID=8444>
- CIBSE. (1999). *CIBSE Environmental Design Guide, Book A*. London: CIBSE.
- Davies, M. G. (2005). *Building heat transfer*. Liverpool: John Wiley & Sons, Ltd .
- DOE. (2008). *International Energy Outlook 2008*. Washington: U.S. DOE.
- Dutch/Shell, R. (2005). *The Shell Global Scenarios to 2025*. Inst. Int. Econ: Washington.
- E.W. Lemmon, M.L. Huber, M.O. McLinden. (2007). *Reference Fluid Thermodynamic and Transport Properties*. USA.
- Effect, T. F. (2008). *The Fuel Effect*. <http://www.thefueleffect.co.uk/images2/savings.gif>
- Gnielinski, V. (1976). New Equations for Heat and Mass Transfer in Turbulent Pipe and Channel Flow. *International Chemical Engineering* 16 , 359-368.
- Harvey, D. (2009). *Energy and the new reality: facing up to climatic change*. Washington: Island Press.
- Hashimoto, K. (2006). Technology and Market Development of CO2 Heat Pump Water Heaters (EcoCute) in Japan. *IEA Heat Pump Centre Newsletter* , 12-16.
- HeatPumpCenter. (2006). *Heat pumps can cut global CO2 emissions by nearly 8%*. Boras: Technical Research Institute of Sweden.
- IEA. (2006). *World energy Outlook 2006*. Paris: IEA.
- IEA. (2008). *Worldwide Trends in Energy Use and Efficiency Key Insights from IEA Indicator Analysis*. Paris: IEA/OECD.
- IPCC. (2007). *Summary for Policymakers. In: Climate Change 2007: The Physical Science Basis. Contribution of Working*. Cambridge: Cambridge University Press,.
- Koppelaar, R. (2005). *World Oil Production & Peaking Outlook*. Amsterdam: Peak Oil Netherlands Foundation.

MatWeb. (16 February 2009). *Electrolytic tough pitch Copper, UNS C11000, OSO25 Temper flat products, tube*: <http://www.matweb.com/search/DataSheet.aspx?MatGUID=ef38978fb59a4887948a717dcbacdfef&ckck=1>

NOAA. (5. January 2009). *Trends in Atmospheric Carbon Dioxide - Mauna Loa*. Trends in Atmospheric Carbon Dioxide - Mauna Loa: www.esrl.noaa.gov/gmd/ccgg/trends

Ochsner, K. (2008). *Geothermal Heat Pumps. A guide for Planning and Installing*. London: Earthscan.

Petukhov, B. (1970). *Heat Transfer and Friction in Turbulent Pipe Flow with Variable Physical Properties*. New York: Academic Press.

Quen Cheer Industrial Co., L. (2009). *Quen Cheer Industrial Co., Ltd.* <http://www.b2btubes.com/product.htm>

Samorka. (2009). *District Heating*. <http://www.samorka.is/Apps/WebObjects/Samorka.woa/1/wa/dp?id=1000059&wosid=7oNY7cmw3J6uzntrWRoB10>

SANYO. (2006). *CO2 NATURAL REFRIGERANT Heat Pump Hot Water Supplier*. Milan: SANYO Airconditioners Europe srl.

SANYO. (2005). *Introduction to our CO2 compressors*. <http://www.sanyo.co.jp/comp-unit/english/co2/pdf/c-cv133.pdf>

Valdimarsson, P. (2008). *District Heating Systems: Fundamentals and applications*. Akureyri.

VDI. (1991). *VDI Heat Atlas*. Wurzberg: VDI Verlag.

WEC. (2008). *World Energy Council Publications*. http://www.worldenergy.org/documents/fig_coal_1_2.gif

Werner Zittel, Jörg Schindler. (2007). *Crude oil - the Supply Outlook*. Ottobrun: Energy Wath Group.

Zukauskas, A. (1987). *Heat Transfer from Tubes in Cross Flow*. New York: Wiley Interscience.

APPENDIX A

5.1 Table A - Properties of CO₂ in temperature range 4°C - 37°C and pressure 38 bar

5.1.1 Part 1

i	Temperature (°C)	Pressure (MPa)	Volume (m ³ /kg)	Enthalpy (kJ/kg)	Therm. Cond. (mW/m ² *K)	Kin. Viscosity (cm ² /s)
1	4	3,8	0,0093	430,16	20,759	0,0014
2	4,5	3,8	0,0094	431,14	20,677	0,0014
3	5	3,8	0,0094	432,1	20,601	0,0014
4	5,5	3,8	0,0095	433,05	20,533	0,0014
5	6	3,8	0,0096	433,99	20,47	0,0015
6	6,5	3,8	0,0096	434,91	20,412	0,0015
7	7	3,8	0,0097	435,82	20,36	0,0015
8	7,5	3,8	0,0097	436,71	20,311	0,0015
9	8	3,8	0,0098	437,6	20,267	0,0015
10	8,5	3,8	0,0099	438,47	20,227	0,0015
11	9	3,8	0,0099	439,33	20,19	0,0015
12	9,5	3,8	0,0100	440,18	20,156	0,0015
13	10	3,8	0,0100	441,02	20,126	0,0015
14	10,5	3,8	0,0101	441,86	20,098	0,0015
15	11	3,8	0,0101	442,68	20,073	0,0016
16	11,5	3,8	0,0102	443,5	20,051	0,0016
17	12	3,8	0,0102	444,31	20,031	0,0016
18	12,5	3,8	0,0103	445,11	20,013	0,0016
19	13	3,8	0,0104	445,9	19,997	0,0016
20	13,5	3,8	0,0104	446,68	19,983	0,0016
21	14	3,8	0,0105	447,46	19,971	0,0016
22	14,5	3,8	0,0105	448,24	19,961	0,0016
23	15	3,8	0,0106	449	19,952	0,0016
24	15,5	3,8	0,0106	449,76	19,945	0,0016
25	16	3,8	0,0107	450,52	19,94	0,0017
26	16,5	3,8	0,0107	451,27	19,936	0,0017
27	17	3,8	0,0108	452,01	19,933	0,0017
28	17,5	3,8	0,0108	452,75	19,931	0,0017
29	18	3,8	0,0109	453,48	19,931	0,0017
30	18,5	3,8	0,0109	454,21	19,932	0,0017
31	19	3,8	0,0109	454,94	19,934	0,0017
32	19,5	3,8	0,0110	455,66	19,937	0,0017
33	20	3,8	0,0110	456,37	19,941	0,0017

34	20,5	3,8	0,0111	457,08	19,947	0,0017
35	21	3,8	0,0111	457,79	19,953	0,0017
36	21,5	3,8	0,0112	458,49	19,96	0,0018
37	22	3,8	0,0112	459,19	19,968	0,0018
38	22,5	3,8	0,0113	459,89	19,976	0,0018
39	23	3,8	0,0113	460,58	19,986	0,0018
40	23,5	3,8	0,0114	461,26	19,996	0,0018
41	24	3,8	0,0114	461,95	20,007	0,0018
42	24,5	3,8	0,0115	462,63	20,019	0,0018
43	25	3,8	0,0115	463,31	20,031	0,0018
44	25,5	3,8	0,0115	463,98	20,044	0,0018
45	26	3,8	0,0116	464,65	20,058	0,0018
46	26,5	3,8	0,0116	465,32	20,072	0,0018
47	27	3,8	0,0117	465,99	20,087	0,0019
48	27,5	3,8	0,0117	466,65	20,103	0,0019
49	28	3,8	0,0118	467,31	20,119	0,0019
50	28,5	3,8	0,0118	467,97	20,135	0,0019
51	29	3,8	0,0118	468,63	20,153	0,0019
52	29,5	3,8	0,0119	469,28	20,17	0,0019
53	30	3,8	0,0119	469,93	20,188	0,0019
54	30,5	3,8	0,0120	470,58	20,207	0,0019
55	31	3,8	0,0120	471,22	20,226	0,0019
56	31,5	3,8	0,0121	471,86	20,245	0,0019
57	32	3,8	0,0121	472,5	20,265	0,0019
58	32,5	3,8	0,0121	473,14	20,286	0,0020
59	33	3,8	0,0122	473,78	20,306	0,0020
60	33,5	3,8	0,0122	474,41	20,327	0,0020
61	34	3,8	0,0123	475,04	20,349	0,0020
62	34,5	3,8	0,0123	475,67	20,371	0,0020
63	35	3,8	0,0123	476,3	20,393	0,0020
64	35,5	3,8	0,0124	476,93	20,416	0,0020
65	36	3,8	0,0124	477,55	20,439	0,0020
66	36,5	3,8	0,0125	478,17	20,462	0,0020
67	37	3,8	0,0125	478,79	20,485	0,0020

5.1.2 Part 2

i	Prandtl	Re	Velocity (m/s)	Nu	h (W/(m ² *°C))	f
1	1,45	27496	5,55	98,50	2921,05	0,0246
2	1,43	27478	5,58	97,79	2888,69	0,0246
3	1,41	27458	5,62	97,12	2858,10	0,0246
4	1,40	27438	5,66	96,47	2829,68	0,0246
5	1,38	27418	5,70	95,85	2802,82	0,0246
6	1,36	27397	5,73	95,25	2777,43	0,0246
7	1,35	27375	5,77	94,67	2753,65	0,0246

8	1,34	27352	5,80	94,12	2730,83	0,0246
9	1,32	27329	5,84	93,58	2709,33	0,0246
10	1,31	27306	5,87	93,06	2689,07	0,0246
11	1,30	27281	5,91	92,56	2669,60	0,0246
12	1,29	27256	5,94	92,07	2651,06	0,0246
13	1,27	27230	5,97	91,59	2633,44	0,0246
14	1,26	27204	6,01	91,13	2616,61	0,0246
15	1,25	27180	6,04	90,69	2600,65	0,0246
16	1,24	27152	6,07	90,25	2585,28	0,0246
17	1,23	27128	6,10	89,84	2570,77	0,0247
18	1,23	27100	6,14	89,42	2556,51	0,0247
19	1,22	27071	6,17	89,01	2542,89	0,0247
20	1,21	27044	6,20	88,62	2529,99	0,0247
21	1,20	27017	6,23	88,24	2517,56	0,0247
22	1,19	26989	6,26	87,87	2505,64	0,0247
23	1,18	26960	6,29	87,50	2494,08	0,0247
24	1,18	26931	6,32	87,14	2482,93	0,0247
25	1,17	26903	6,35	86,79	2472,37	0,0247
26	1,16	26872	6,38	86,45	2461,98	0,0247
27	1,16	26843	6,41	86,11	2452,06	0,0247
28	1,15	26814	6,44	85,78	2442,50	0,0247
29	1,14	26786	6,47	85,46	2433,41	0,0247
30	1,14	26756	6,49	85,15	2424,50	0,0247
31	1,13	26723	6,52	84,83	2415,65	0,0247
32	1,12	26695	6,55	84,53	2407,56	0,0248
33	1,12	26665	6,58	84,23	2399,43	0,0248
34	1,11	26635	6,61	83,94	2391,88	0,0248
35	1,11	26603	6,63	83,64	2384,17	0,0248
36	1,10	26575	6,66	83,37	2377,09	0,0248
37	1,10	26541	6,69	83,08	2369,92	0,0248
38	1,09	26512	6,72	82,81	2363,15	0,0248
39	1,09	26483	6,74	82,54	2356,74	0,0248
40	1,08	26451	6,77	82,28	2350,29	0,0248
41	1,08	26421	6,80	82,01	2344,07	0,0248
42	1,07	26388	6,82	81,75	2337,93	0,0248
43	1,07	26357	6,85	81,50	2332,20	0,0248
44	1,06	26327	6,88	81,25	2326,52	0,0248
45	1,06	26295	6,90	81,00	2321,11	0,0248
46	1,06	26264	6,93	80,76	2315,80	0,0249
47	1,05	26232	6,95	80,52	2310,57	0,0249
48	1,05	26200	6,98	80,28	2305,58	0,0249
49	1,04	26171	7,01	80,05	2300,87	0,0249
50	1,04	26138	7,03	79,82	2295,98	0,0249
51	1,04	26107	7,06	79,60	2291,59	0,0249
52	1,03	26076	7,08	79,37	2287,09	0,0249
53	1,03	26044	7,11	79,15	2282,69	0,0249

54	1,03	26011	7,13	78,93	2278,52	0,0249
55	1,02	25981	7,16	78,72	2274,57	0,0249
56	1,02	25949	7,18	78,51	2270,53	0,0249
57	1,02	25917	7,21	78,30	2266,68	0,0249
58	1,01	25886	7,23	78,09	2263,00	0,0249
59	1,01	25854	7,26	77,88	2259,28	0,0250
60	1,01	25822	7,28	77,68	2255,70	0,0250
61	1,00	25791	7,31	77,48	2252,34	0,0250
62	1,00	25760	7,33	77,28	2249,09	0,0250
63	1,00	25726	7,35	77,08	2245,67	0,0250
64	0,99	25696	7,38	76,89	2242,65	0,0250
65	0,99	25664	7,40	76,70	2239,57	0,0250
66	0,99	25633	7,43	76,51	2236,55	0,0250
67	0,99	25600	7,45	76,32	2233,51	0,0250

5.2 Table B - Properties of water in temperature range 15°C - 40°C and pressure 2 bar

5.2.1 Part 1

j	Temperature (°C)	Pressure (bar)	Density (kg/m ³)	Volume (m ³ /kg)	Enthalpy (kJ/kg)	Re number	Velocity (m/s)
1	7	2	999,95	0,001	29,62	19,16	0,014
2	7,5	2	999,93	0,001	31,72	19,45	0,014
3	8	2	999,9	0,001	33,82	19,75	0,014
4	8,5	2	999,87	0,001	35,92	20,04	0,014
5	9	2	999,83	0,001	38,02	20,34	0,014
6	9,5	2	999,79	0,001	40,12	20,64	0,014
7	10	2	999,75	0,001	42,22	20,94	0,014
8	10,5	2	999,7	0,001	44,31	21,24	0,014
9	11	2	999,65	0,001	46,41	21,54	0,014
10	11,5	2	999,6	0,001	48,51	21,85	0,014
11	12	2	999,55	0,001001	50,60	22,16	0,014
12	12,5	2	999,49	0,001001	52,70	22,47	0,014
13	13	2	999,43	0,001001	54,79	22,78	0,014
14	13,5	2	999,36	0,001001	56,89	23,09	0,014
15	14	2	999,29	0,001001	58,98	23,40	0,014
16	14,5	2	999,22	0,001001	61,08	23,72	0,014
17	15	2	999,15	0,001001	63,17	24,04	0,014
18	15,5	2	999,07	0,001001	65,27	24,35	0,014
19	16	2	998,99	0,001001	67,36	24,68	0,014
20	16,5	2	998,91	0,001001	69,45	25,00	0,014
21	17	2	998,82	0,001001	71,55	25,32	0,014

22	17,5	2	998,74	0,001001	73,64	25,65	0,014
23	18	2	998,64	0,001001	75,73	25,98	0,014
24	18,5	2	998,55	0,001002	77,82	26,30	0,014
25	19	2	998,45	0,001002	79,92	26,63	0,014
26	19,5	2	998,35	0,001002	82,01	26,96	0,014
27	20	2	998,25	0,001002	84,10	27,30	0,014
28	20,5	2	998,15	0,001002	86,19	27,63	0,014
29	21	2	998,04	0,001002	88,28	27,97	0,014
30	21,5	2	997,93	0,001002	90,38	28,31	0,014
31	22	2	997,82	0,001002	92,47	28,65	0,014
32	22,5	2	997,7	0,001002	94,56	28,99	0,014
33	23	2	997,59	0,001002	96,65	29,33	0,014
34	23,5	2	997,47	0,001003	98,74	29,67	0,014
35	24	2	997,34	0,001003	100,83	30,02	0,014
36	24,5	2	997,22	0,001003	102,92	30,37	0,014
37	25	2	997,09	0,001003	105,01	30,72	0,014
38	25,5	2	996,96	0,001003	107,10	31,07	0,014
39	26	2	996,83	0,001003	109,19	31,42	0,014
40	26,5	2	996,7	0,001003	111,28	31,77	0,014
41	27	2	996,56	0,001004	113,37	32,13	0,014
42	27,5	2	996,42	0,001004	115,46	32,48	0,014
43	28	2	996,28	0,001004	117,55	32,84	0,014
44	28,5	2	996,14	0,001004	119,64	33,20	0,014
45	29	2	995,99	0,001004	121,73	33,56	0,014
46	29,5	2	995,84	0,001004	123,82	33,93	0,014
47	30	2	995,69	0,001004	125,91	34,29	0,014
48	30,5	2	995,54	0,001005	128,00	34,66	0,014
49	31	2	995,39	0,001005	130,09	35,02	0,014
50	31,5	2	995,23	0,001005	132,18	35,39	0,014
51	32	2	995,07	0,001005	134,27	35,76	0,014
52	32,5	2	994,91	0,001005	136,36	36,13	0,014
53	33	2	994,75	0,001005	138,45	36,50	0,014
54	33,5	2	994,58	0,001005	140,54	36,88	0,014
55	34	2	994,42	0,001006	142,63	37,25	0,014
56	34,5	2	994,25	0,001006	144,72	37,63	0,014
57	35	2	994,08	0,001006	146,81	38,01	0,014
58	35,5	2	993,9	0,001006	148,90	38,39	0,014
59	36	2	993,73	0,001006	150,99	38,77	0,014
60	36,5	2	993,55	0,001007	153,08	39,15	0,014
61	37	2	993,37	0,001007	155,17	39,54	0,014
62	37,5	2	993,19	0,001007	157,26	39,92	0,014
63	38	2	993,01	0,001007	159,35	40,31	0,014
64	38,5	2	992,83	0,001007	161,44	40,70	0,014
65	39	2	992,64	0,001007	163,52	41,09	0,014
66	39,5	2	992,45	0,001008	165,61	41,48	0,014
67	40	2	992,26	0,001008	167,70	41,87	0,014

5.2.2 Part 2

j	Therm. Cond. (mW/(m*K))	Vmax (m/s)	Kin. Viscosity (cm ² /s)	Prandtl	h (W/(m ² *°C))	Nu
1	574,42	0,023	0,014	10,43	3188,68	6,66
2	575,37	0,023	0,014	10,26	3180,36	6,63
3	576,32	0,023	0,014	10,09	3171,88	6,60
4	577,26	0,023	0,014	9,92	3163,58	6,58
5	578,21	0,023	0,013	9,76	3155,46	6,55
6	579,15	0,023	0,013	9,60	3147,34	6,52
7	580,1	0,023	0,013	9,44	3139,55	6,49
8	581,04	0,023	0,013	9,29	3131,67	6,47
9	581,98	0,023	0,013	9,14	3123,90	6,44
10	582,91	0,023	0,013	9,00	3116,32	6,42
11	583,85	0,023	0,012	8,86	3108,91	6,39
12	584,79	0,023	0,012	8,72	3101,46	6,36
13	585,72	0,023	0,012	8,59	3094,26	6,34
14	586,65	0,023	0,012	8,46	3086,89	6,31
15	587,58	0,023	0,012	8,33	3079,82	6,29
16	588,5	0,023	0,012	8,20	3072,82	6,27
17	589,43	0,023	0,011	8,08	3066,02	6,24
18	590,35	0,023	0,011	7,96	3059,05	6,22
19	591,27	0,023	0,011	7,85	3052,31	6,19
20	592,18	0,023	0,011	7,73	3045,62	6,17
21	593,09	0,023	0,011	7,62	3039,00	6,15
22	594	0,023	0,011	7,51	3032,48	6,13
23	594,91	0,023	0,011	7,41	3026,13	6,10
24	595,82	0,023	0,010	7,30	3019,70	6,08
25	596,72	0,023	0,010	7,20	3013,29	6,06
26	597,61	0,023	0,010	7,10	3007,08	6,04
27	598,51	0,023	0,010	7,00	3001,07	6,02
28	599,4	0,023	0,010	6,91	2994,95	6,00
29	600,28	0,023	0,010	6,81	2988,88	5,97
30	601,17	0,023	0,010	6,72	2982,97	5,95
31	602,05	0,023	0,010	6,63	2977,04	5,93
32	602,92	0,023	0,009	6,54	2971,19	5,91
33	603,79	0,023	0,009	6,46	2965,39	5,89
34	604,66	0,023	0,009	6,37	2959,65	5,87
35	605,53	0,023	0,009	6,29	2954,13	5,85
36	606,38	0,023	0,009	6,21	2948,46	5,83
37	607,24	0,023	0,009	6,13	2942,91	5,82
38	608,09	0,023	0,009	6,05	2937,38	5,80
39	608,94	0,023	0,009	5,97	2932,06	5,78
40	609,78	0,023	0,009	5,90	2926,63	5,76
41	610,62	0,023	0,009	5,83	2921,38	5,74
42	611,45	0,023	0,008	5,75	2916,04	5,72

43	612,28	0,023	0,008	5,68	2910,78	5,70
44	613,1	0,023	0,008	5,61	2905,64	5,69
45	613,92	0,023	0,008	5,55	2900,47	5,67
46	614,74	0,023	0,008	5,48	2895,46	5,65
47	615,55	0,023	0,008	5,41	2890,37	5,63
48	616,35	0,023	0,008	5,35	2885,40	5,62
49	617,15	0,023	0,008	5,29	2880,39	5,60
50	617,95	0,023	0,008	5,22	2875,56	5,58
51	618,74	0,023	0,008	5,16	2870,71	5,57
52	619,52	0,023	0,008	5,10	2865,78	5,55
53	620,3	0,023	0,008	5,05	2861,05	5,53
54	621,08	0,023	0,007	4,99	2856,24	5,52
55	621,85	0,0229	0,007	4,93	2851,59	5,50
56	622,61	0,0229	0,007	4,88	2846,94	5,49
57	623,37	0,0229	0,007	4,82	2842,33	5,47
58	624,12	0,0229	0,007	4,77	2837,62	5,46
59	624,87	0,0229	0,007	4,72	2833,11	5,44
60	625,62	0,0229	0,007	4,66	2828,61	5,43
61	626,36	0,0229	0,007	4,61	2824,18	5,41
62	627,09	0,0229	0,007	4,56	2819,71	5,40
63	627,82	0,0229	0,007	4,51	2815,21	5,38
64	628,54	0,0229	0,007	4,47	2810,81	5,37
65	629,25	0,023	0,007	4,42	2806,46	5,35
66	629,97	0,023	0,007	4,37	2802,16	5,34
67	630,67	0,023	0,007	4,33	2797,84	5,32

6 APPENDIX B

This appendix will explain how to estimate heat demand for a small building.

6.1 Approach

The simplest and most accurate approach is to assume steady state conditions, design temperature, house temperature and surface conductance (inside and outside a building). The rest are calculations concerned with the geometry of the building and properties of the materials.

Energy balance for a building can be written as:

$$\dot{Q}_{in} - \dot{Q}_{out} = \frac{dE_{walls}}{dt}$$

Where $\dot{Q}_{in}$ is power of a heat source, $\dot{Q}_{out}$ are heat losses and $\frac{dE_{walls}}{dt}$ is the rate of change of the energy of the walls. A steady state in energy balance means that there are no temperature changes in the walls at any point in time, so $\frac{dE_{walls}}{dt} = 0$ and:

$$\dot{Q}_{in} = \dot{Q}_{out}$$

This means that in order to choose the power of the heat source we need to calculate building heat losses.

$\dot{Q}_{out}$ is a sum of:

- heat losses through many different elements of the building - $\dot{Q}_A$
- ventilation losses - $\dot{Q}_{ven}$

$\dot{Q}_A$ is also a sum of a few heat flows. They are divided into three groups:

- through windows and doors,
- walls and roofs,
- balconies

All losses through the elements of the building can be calculated in a similar way because they depend on: temperature difference $\Delta T = (T_{in} - T_{out})$, surface conductance inside (h_{in}) and outside (h_{out}) the building, material properties (thermal conductivity - k) and geometry of the elements.

6.2 Assumptions

It is impossible to calculate heat demand for a building with 100% accuracy, because too many factors affect the final value: number of inhabitants, electrical equipment, permeability of windows or doors, construction defects and so on. In this chapter the most important elements which will allow us to estimate, quite quickly, heat demand for a small building will be presented.

The following assumptions were made:

- heat flow through walls, windows and doors is one dimensional
- heat flow through balconies is two dimensional
- balconies are modeled as fins with isolated tip
- walls are calculated with the use of thermal resistance concept

6.3 Design temperature

To start calculations we need to assume the temperature inside the house – T_{in} (which is about 20°C) and ambient temperature T_{out} . It is our design temperature and it is different at every location. The following example will be used to demonstrate how to obtain it:

In Reykjavik temperature during heating season is almost all the time above -8°C, except 5 days (not in row) when it reaches even -17°C (figure 38). It is not practical to project heating installation for ambient temperature -17°C, since it lasts continuously for a few hours or a day and such an installation needs much more power (which will be fully used on only 5 days a year). Additionally, it is senseless to assume -17°C, because of building heat capacity which can delay cooling for 20-40 hours (Valdimarsson, 2008) (which is usually sufficient). It is much better to assume design temperature to be -8°C. This approach allows us to save energy without any harm done to thermal comfort.

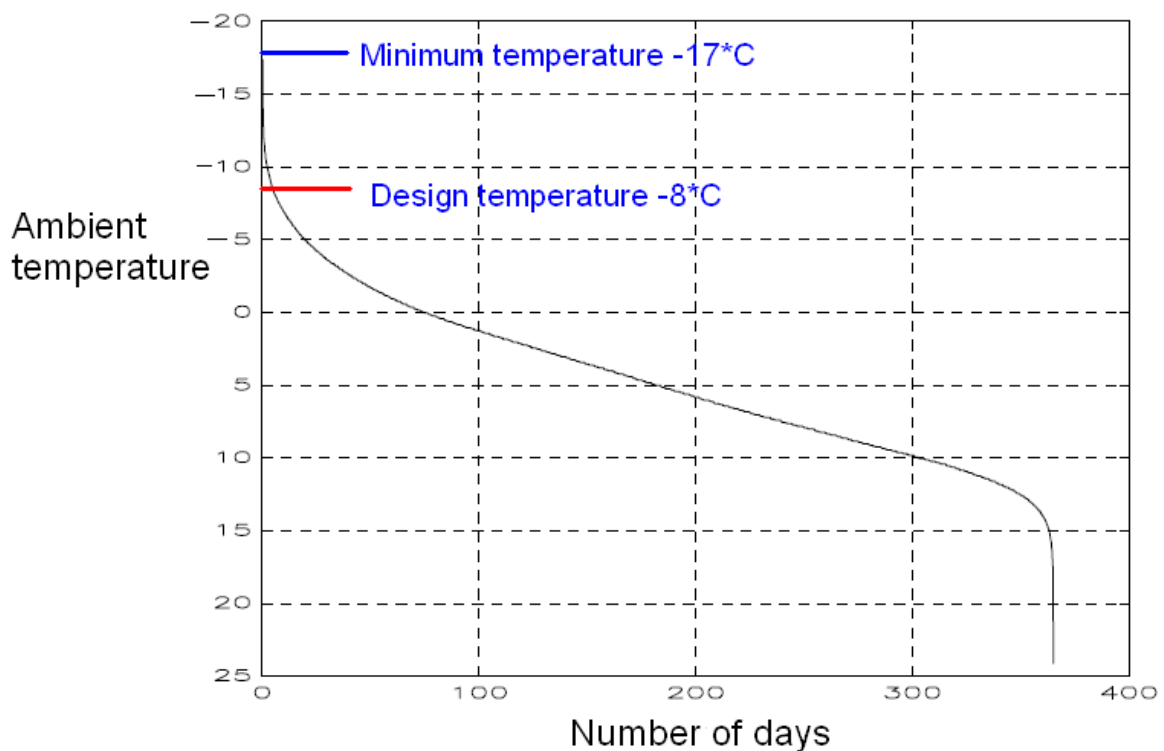


Figure 38 Temperatures in Reykjavik during one year, sorted in a decreasing order (Valdimarsson, 2008)

6.3.1 Surface conductances

Other parameters which need to be assumed are surface conductances, which are combined convection and radiation heat transfer coefficients. They are estimated experimentally, some popular values are listed in the figure 39.

Position	Direction of heat flow	h for surface emittance		
		0,9	0,2	0,05
Stagnant air (both indoors and outdoors)				
Horizontal	Up	9,26	5,17	4,32
Horizontal	Down	6,13	2,10	1,25
45° slope	Up	9,09	5,00	4,15
45° slope	Down	7,50	3,41	2,56
Vertical	Horizontal	8,29	4,20	3,35
Moving air (any position, any direction)				
Winter condition (winds 24 km/h)		34,00	-	-
Summer conditions (winds 12 km/h)		22,70	-	-

Figure 39 Surface conductances (ASHRAE, 1993)

6.4 Losses through doors and windows

The easiest to calculate are the losses through windows and doors. It is because producers always give U-value - parameter which tells the conductivity of the material from which windows (or doors) were made of. To calculate heat losses through one type of windows or doors we use the equation below:

$$\dot{Q}_{loss} = n \times A \times U \times \Delta T$$

Where n is a number of elements of a given type, A is its area, and ΔT is the temperature difference between the temperature inside the house and the design temperature. To obtain total losses through windows and doors we need to find the sum of heat losses for different types.

6.5 Losses through walls

Losses through walls are calculated with use of the thermal resistance concept. Thermal resistance R tells us what the heat flow between two temperatures will be. In our case, between T_{in} and T_{out} , this means that heat flow (loss) through one type of a wall will be described by the equation below:

$$\dot{Q}_{loss\ wall} = \frac{T_{in} - T_{out}}{R_T}$$

Where R_T is total resistance and it consists of the resistance of thermal resistances against convection inside the house $R_{conv, in}$, outside $R_{conv, out}$ and wall resistance R_{wall} .

$$R_{conv,in} = \frac{1}{h_{in} \times A} \text{ and } R_{conv,out} = \frac{1}{h_{out} \times A}$$

Resistance of the wall:

$$R_{wall} = \frac{L}{k \times A}$$

Where L is the length of the wall, k is thermal conductivity and A is area of the wall.

If the wall consists of layers connected in a series, then we add resistances of the layers. If the wall consists of layers connected in parallel then we add reciprocals of resistances. Figure 40 explains it in an example with two layers. If we deal with a combined series-parallel arrangement we should calculate resistances for individual elements and then determine total heat transfer and total resistance using relations from the figure 41.

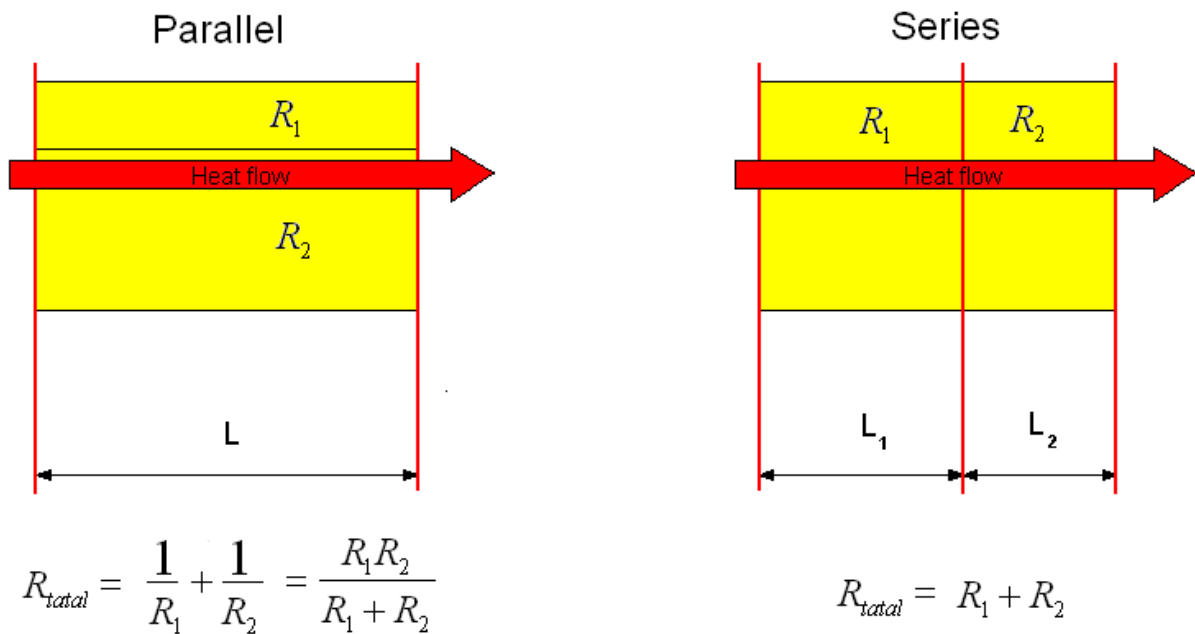
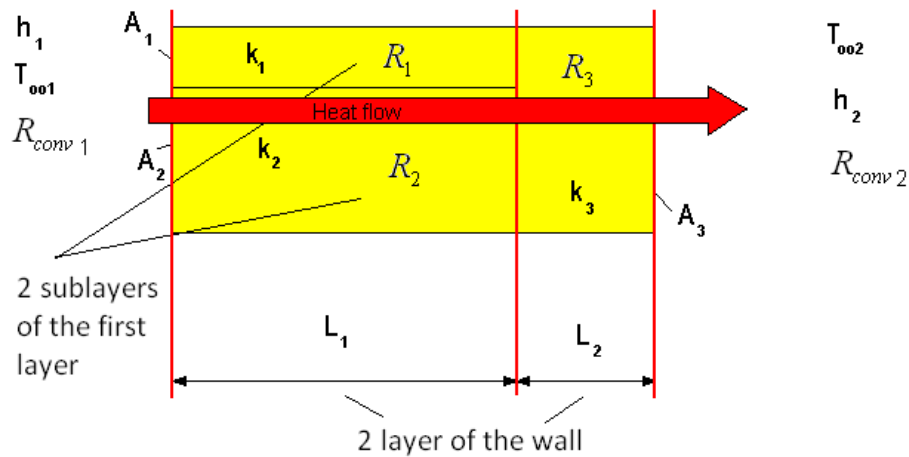


Figure 40 Parallel and in series thermal resistances arrangement



$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} \quad R_{total} = \frac{R_1 R_2}{R_1 + R_2} + R_3 + R_{conv 1} + R_{conv 2}$$

$$R_i = \frac{L_i}{k_i A_i} \quad R_{conv 1} = \frac{1}{h_1 (A_1 + A_2)} \quad R_{conv 2} = \frac{1}{h_2 A_3}$$

$i = 1, 2, 3$

Figure 41 An example of parallel-series arrangement

6.6 Losses through balconies

Balconies can be modeled as fins. To know heat losses from a fin we need to solve the fin equation, which is a second order differential equation. With no boundary conditions at the end of the fin it is impossible (at the beginning temperature is known and equal to wall house temperature T_{in}), but if we assume that the fin has an isolated tip, then the equation simplifies to:

$$\dot{Q}_{balcony} = \sqrt{A_c h_{out} p k} (T_{in} - T_{out}) \tanh aL$$

$$a = \sqrt{\frac{h_{out} p}{k A_c}}$$

p – perimeter of the fin

k – thermal conductivity of the fin's material

A_c – cross-sectional area of the fin

L – length of the fin

Losses through the fin's tip are usually negligible compared to all losses from a fin, so it is quite a good approximation.

6.7 Ventilation losses

Ventilation losses are responsible for 25-50% of total demand for energy for space heating purpose (Davies, 2005). They can be approximately calculated from the equation below (for one room):

$$\dot{Q}_{ven} = \frac{1}{3}NV_{room}(T_{in} - T_{out})$$

V_{room} - room volume [m^3]

N – number of air changes per hour, value usually between 0.5 to 1 (CIBSE, 1999)

To obtain total ventilation losses we should sum values for all rooms in the house.



7 APPENDIX C

To help with the heat exchanger design, an application in Visual Basic 2008 was created, for a case in which some of previous assumptions contained in the thesis will have to be changed. The same limitations and equations, as in the main part of the thesis are assumed. The program calculates the length of the evaporator, the length of the superheater, pressure losses, COP and the water temperature at the outlet.

It has limits for input data, but an end-user should know that to receive realistic output, input should be also realistic.

The figure displays two screenshots of a Visual Basic application titled 'Appendix C'. The left window shows the 'Input' tab, and the right window shows the 'Output' tab.

Input Tab (Left Window):

Please insert data:

Parameter	Value
Water temperature at the inlet to the evaporator [°C]	33
Internal tube diameter [mm]	0,7
External tube diameter [mm]	1,2
Thermal conductivity of the tube [W/(m·K)]	400
Water mass flow [kg/s]	0,033
Water duct height - h [mm]	20
Water duct width - l [mm]	100
Distance between tubes - St [mm]	3
Number of tubes	64

Calculate

Output Tab (Right Window):

Parameter	Value
Outlet water temperature [°C]	6,456985
COP	5,393892
Length of the heat exchanger [m]	0,9433383
Pressure losses [bar]	1,33323
Heat exchanged in the superheater [kW]	0,6149872
Heat exchanged in the evaporator [kW]	3,050736
Total heat exchanged [kW]	3,665723
Length of the superheater [m]	0,2656755
Length of the evaporator [m]	0,6776628

Figure 42 Program for calculations of the heat exchanger